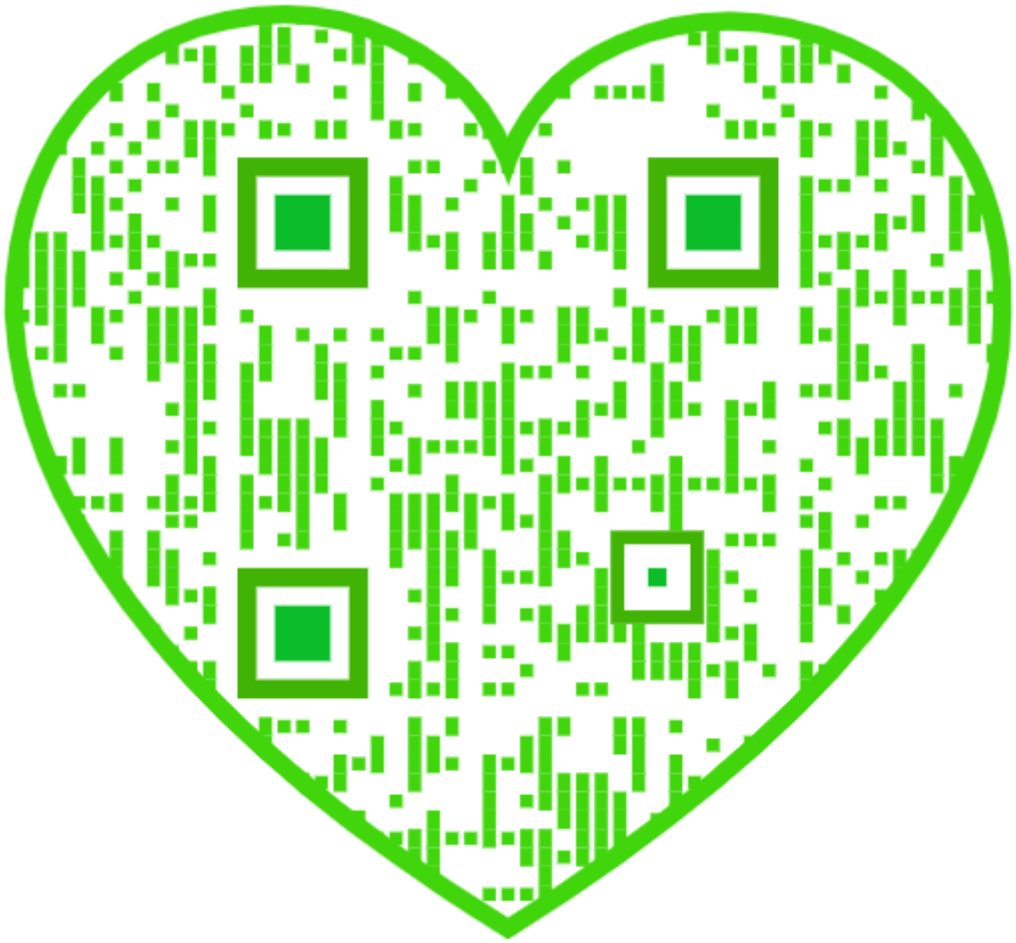
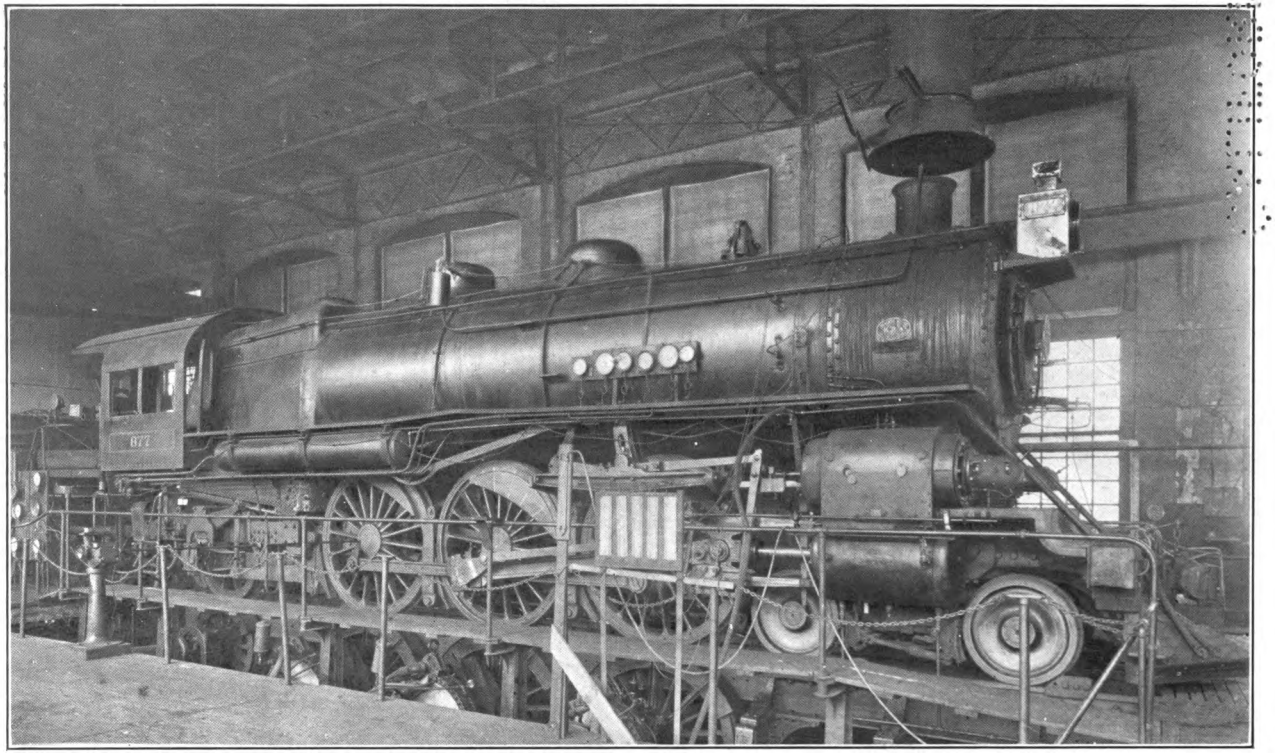


LOCOMOTIVE OPERATION
AND
TRAIN CONTROL



A MODERN PACIFIC TYPE LOCOMOTIVE, EQUIPPED FOR TESTS ON LOCOMOTIVE TESTING PLANT.

(*Frontispiece.*)

PRINCIPLES OF
LOCOMOTIVE OPERATION
AND
TRAIN CONTROL

BY

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UNIV. OF CALIFORNIA

McGRAW-HILL BOOK COMPANY, Inc.
239 WEST 39TH STREET, NEW YORK
6 BOUVERIE STREET, LONDON, E. C.
1915

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PREFACE

The steam railroad is the greatest commercial interest in the country. The adoption of steel cars and heavier trains, running at average high speeds, has introduced new and exacting problems. Hemmed in by the limitations of road clearances and allowable weights, the mechanical engineer has produced the "modern locomotive," which is a well-equipped and efficient power plant of two thousand or more horse power. This achievement has been reached only by a scientific study of locomotive operation.

The absence of a text-book presenting the recent developments in locomotive performance and including a study of air brakes, has led the author to prepare this elementary treatise. While written primarily for use in technical schools, the engineer may here review the theory on which are based certain problems in design and construction. The author has presented the principles, beyond which lies the field for extended study.

Theory is, in the main, our safest guide to trustworthy results. How true this was in the adoption of superheated steam for locomotives! Before its application, theory anticipated to a nicety the results which later were obtained in operation. This applies also to the practical applications of "speed-time" curves, as set forth in this volume. However, theory alone has its limitations and the locomotive testing plant has done much to stimulate new interest in the study of locomotive performance.

Material has been used from many sources and due credit is given in the text to the various writers and publications. Acknowledgment is also made for timely suggestions from engineers of the Test Department, Pennsylvania Railroad, and for assistance from Mr. J. T. Wallis, Gen. Supt. Motive Power, Pennsylvania Railroad, Mr. F. H. Parke, Resident Engineer, Westinghouse Air Brake Co., Mr. W. F. Kiesel, Asst. Mech. Engr., and Mr. C. E. Barba, Asst. Engr., Pennsylvania Railroad. The author is indebted to Mr. Geo. Rhodes for carefully reading part of the proof.

A. J. W.

THE PENNSYLVANIA STATE COLLEGE,
STATE COLLEGE, PA.,
September, 1915

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LOCOMOTIVE OPERATION

CHAPTER I

FIRST PRINCIPLES

The Locomotive Problem Stated; Essential Parts of a Locomotive; Locomotive a Power Plant

Work.—The purpose of a locomotive is to do work. Work is overcoming resistance through a space and is, therefore, the product of two quantities, (1) a *force* and (2) a *distance* through which that force acts.

When a force is measured in pounds and the distance in feet, the unit of work is a foot-pound. If 10 lb. rest on a smooth surface and it takes 2 lb. pull to overcome frictional resistance and to keep it moving on the surface, then the work done in moving it 10 ft. will be $10 \times 2 = 20$ ft.-lb. If, however, the weight is raised 10 ft., the work will be $10 \times 10 = 100$ ft.-lb. Where the metric system is used, the mechanical work is measured by the product of the force in kilograms (2.2046 lb.) and the distance in meters (3.28 ft.). The product is in kilogrammeters (7.233 ft.-lb.).

When steam forces a piston to move against a resistance, work is done; but if the steam pushes against a piston and the force thereby exerted is not sufficient to overcome the opposing resistances, no work will be performed. If a piston is pushed through the entire length of a stroke with a total force of 100,000 lb., in 1 sec., the same amount of work will be done as if it took one-fifth of a second for the same piston travel, but the *power* will be five times as much in the latter case, since *power is a rate of doing work.*

Power.—Let us suppose that 20,000 lb. pull is required to move a certain train at a uniform speed of say 5 miles an hour. The work done in moving it the 5 miles will be $20{,}000 \times 5280 \times 5 = 528{,}000{,}000$ ft.-lb., and this will be performed in 1 hr. The horse power required is $528{,}000{,}000 \div (33{,}000 \times 60) = 267$, since 1 hp. equals 33,000 ft.-lb. of work done in 1 min. or 33,000

× 60 ft.-lb. in 1 hr. To do this same amount of work in say 12 min. (one-fifth of an hour) requires 267 × 5 = 1335 hp.

Heat and Work.—Energy is the ability or capacity to do work. Heat is a form of energy, so that when coal burns in the fire-box of a locomotive, we may express its heat energy in terms of useful work. The unit relation between heat and work may be illustrated as follows: Suppose that a pound weight is attached to a string which passes over a pulley at the top of a tower 778 ft. from the ground, and that the other end of the string is fastened to a pulley on an axle which revolves a propeller placed in a perfectly insulated bucket containing 1 lb. of water. Assuming that as the weight falls there is no loss of energy due to friction, then will the temperature of the water be raised 1°F., and the heat required to raise the temperature of 1 lb. of the water through 1°F. is a unit of heat, called a British Thermal Unit. The "Mechanical Equivalent of Heat" is therefore 778 ft.-lb.

Going back to the case of the locomotive exerting 1335 hp., we may now write, 1335 × 33,000 ÷ 778 = 56,630 B.t.u. per min. or approximately equal to the heat evolved in burning 4.5 lb. of coal. It is found that from the heat of a pound of coal burned in the fire box of a locomotive, but about one-twentieth appears as useful work at the track, so that it would require 20 × 4.5 = 90 lb. coal burned to exert the constant pull of 20,000 lb. during every minute.

The foregoing principles apply, in general, to any heat engine. Their particular application to the locomotive may be brought out by the following simple statement of the transformation of heat energy in coal to the mechanical energy made available in hauling a train.

Coal, containing from 12,000 to 15,000 B.t.u. per lb., is delivered by hand or by mechanical stoker to the fire-box, where it first becomes heated, driving off the combustible and other gases. The oxygen required to support combustion, is supplied by air drawn through the grates and hot-fuel bed When the fire-door is opened, additional air passes directly to the combustion chamber. The gases burn at a high temperature, giving up to the water surrounding the fire-box and fire-tubes some 50 to 70 per cent. of the total heat in the coal. These hot gases entering the boiler tubes at a temperature of about 2000°F., pass out at the front end at 500° to 800°F., the required draft being obtained by the back pressure of the exhaust steam from the cylinders. The

exhaust passes up through a nozzle and is discharged at the stack with the burnt gases.

Water, at about 200°F. below the temperature corresponding to the usual boiler pressure, is forced by means of a steam injector into the boiler, where it is heated to boiling temperature and is evaporated. The steam is collected in a dome, marked 6 in Fig. 5, from which it is admitted (through a throttle valve controlled from the cab) into a steam pipe placed above the water level in the boiler. Reaching the front end of the locomotive, the steam is led through branch pipes to each of the cylinders, admission to which is controlled by a valve and valve gear operated by the running gear and controlled by the reverse lever. The steam exhausts through the stack producing the draft, which increases with the effort of the locomotive.

When a superheater is used, about one-fourth of the heating surface is replaced by flues which are 5 to 5½ in. in diameter, through which pass the superheater elements or tubes, the saturated steam being conducted from a header in the front end through these superheater elements where the steam is raised in temperature at constant pressure and is led to a second header from which it passes to the cylinders. Steam entering the superheater tubes, is at 388°F., if the gage pressure is 200 lb. The gases surrounding the tubes are at about 1600°F. at the fire-box end and 700° or 800° at the front end, the steam absorbing an amount of heat sufficient to raise its temperature (at the 200 lb. pressure) to 150° or more above the 388°.

The energy stored as heat in the coal is thus transformed into available energy in the steam. The pressure of the steam on the pistons tends to turn the driving wheels about their axles and, because of this effort, to slip or skid the drivers on the rails; but if the drivers do not slip, the pressure is exerted to produce a "drawbar pull," in turn acting to haul the load.

An electric locomotive derives its power from boiler, engines, and generators placed in a power plant some distance from the train. In the case of a steam locomotive, all of the equipment necessary to develop the power is a part of the locomotive itself. It is consistent, therefore, to consider the steam locomotive as a power plant on wheels.

It has been tersely stated that in a steam locomotive it is necessary to conserve all the heat possible while in an electric locomotive it is necessary to dissipate all the heat possible.

The Steam Locomotive a Power Plant.—Considered as a power plant, there are four distinct divisions to a treatment of the subject:

A.—Fuel and Combustion.
Deals with questions of fire-box, grate and front end.
B.—Steam.
Deals with the questions of the evaporating surface in the boiler.
C.—Utilization of Steam.
Deals with questions of cylinder and of valve gear.
D.—Adhesion.
Deals with questions of the driving mechanism.

The Essential Parts.—The four essential interdependent parts of a locomotive are the *fire-box*, the *boiler*, the *cylinders* and the *driving wheels*. The fire-box capacity is limited by the allowable grate area and the amount of coal which can be burned on that grate. The boiler capacity is limited by the amount of steam which can be produced in a boiler of admissible size and weight. The cylinder merely transmits the power, so that the cylinder power may be indefinitely large, but not less than that required to move the locomotive. The tractive force, or pull at the track by the driving wheels, is limited by the part of the total weight of the locomotive which can be placed on the driving wheels.

Consider, as follows, the dependence of one upon the other of these essential parts: If the boiler or the furnace power is smallest, the steam pressure will fall and the wheels will stop. If the cylinder power is smallest the engine will be stalled when using full steam pressure. If the tractive force is smallest, the engine will slip the drivers. With the drivers, we are concerned with the problems of allowable tractive effort without slipping; with the cylinders, we consider primarily the work done by the steam on the piston; with the boiler sufficient steam must be evaporated at the desired pressure to supply the required force at the track, the cylinder acting merely to transmit the force; with the fire-box, sufficient heat must be supplied to evaporate all the water which can be used to advantage by the boiler.

CHAPTER II

CLASSIFICATION OF LOCOMOTIVES AND ESSENTIAL FEATURES OF PRINCIPAL TYPES

Several methods of classifying locomotives are in common use. Usually, the classification is based on the wheel arrangement of the locomotive, the service in which it is engaged, the size of the cylinders or some combination of all three of the foregoing. Table I is presented to accompany the following explanation.

The method generally adopted uses the Whyte symbols, based solely on wheel arrangement. An American type locomotive having a four-wheel front truck, four coupled driving wheels and no trailing truck is designated under this method as a 4–4–0 type.

In the system developed by the Baldwin Locomotive Works, the wheel arrangement of the locomotive is denoted by an arbitrary system of letters. The total number of wheels under the locomotive is indicated by a number, and the size of the cylinder is determined by dividing the classification number by two and adding three to the remainder. When there are trucks at both ends of the locomotive, the fraction ¼ is placed after the cylinder number and when there is a truck at the rear end and none at the front, the fraction is ⅓. Thus, the symbol 12–44–¼–D–50 represents a locomotive with twelve wheels, cylinders of 25 in. in diameter $\left(\frac{44}{2}+3 = 25\right)$, a truck at each end as shown by the use of the fraction ¼, three pairs of driving wheels (as shown in Table I) and the fiftieth of its class to be built. However, the last number in the above symbol is not usually given.

The Pennsylvania Railroad system assigns a letter to each class of locomotive, based on the wheel arrangement, using a primary letter for a given class, adding a number to designate the different types of any class and a small suffix letter for some modification of the type. The suffix letter "s" is used to denote the addition of a superheater. A certain type of "Pacific" locomotive would

be designated by this system as a K2sa. A number of railroads in the United States have adopted this system or one similar to it, but usually refer to the above as a Class rather than a Type, the latter being designated by the Whyte symbol.

TABLE I.—CLASSIFICATION OF SOME PREVAILING TYPES

Wheel arrangement	Type	Whyte symbol	Baldwin symbol	P.R.R. symbol
◿□OO	4-Wheeled switcher	0–4–0	4–C	A
◿□OOO	6-Wheeled switcher	0–6–0	6–D	B
◿□OOOO	8-Wheeled switcher	0–8–0	8–E	C
◿o□OOO	Mogul	2–6–0	8–D	F
◿o□OOOo	Prairie	2–6–2	10¼–D	J
◿o□OOOO	Consolidation	2–8–0	10–E	H
◿o□OOOOo	Mikado	2–8–2	12¼–E	L
◿o□OOOOO	Decapod	2–10–0	12–F	
◿o□oOO	American	4–4–0	8–C	D
◿o□oOOo	Atlantic	4–4–2	10¼–C	E
◿o□oOOO	10–Wheeled	4–6–0	10–D	G
◿o□oOOOo	Pacific	4–6–2	12¼–D	K
◿o□oOOOO	12–Wheeled	4–8–0	12–E	
◿o□OOOO□OOOOo	Articulated	2–8–8–2	20¼–EE	H–H

NOTE.—The small rectangle representing the cylinder is usually omitted when illustrating the wheel arrangement.

PRINCIPAL TYPES

The selection of a type best adapted for service on a particular roadbed is largely a matter of judgment based on experience, for the factors entering into the question are many and varied. In general, the poorest roadbed for the maximum haul requires the longest wheel base.

1. American Type (4–4–0), Fig. 1.—This type of locomotive has a four-wheel front truck, four coupled driving wheels and no trailing wheels. It was among the first developed by American locomotive builders and from the inception of railroading up to about 1875 was used in all classes of service. Since that time, however, it has been superseded by other types for all work except the handling of passenger trains and within the last few years has been displaced to a large degree for this work by

locomotives of the Atlantic type; so that to-day the American type is used only for light passenger service.

2. Atlantic Type (4–4–2), Fig. 2.—A locomotive having a four-wheel front truck, four coupled driving wheels and a two-wheel trailing truck. This type is a development of the 4–4–0 type, virtually a re-design of that type in which the driving wheels are moved forward under the waist of the boiler ahead of the fire-box, the main rods being connected to the second pair of closely coupled driving wheels, and a pair of trailing wheels added to support the overhanging back end, resulting in a locomotive with a wider fire-box and a boiler of larger diameter with much greater steaming capacity. The 4–4–2 type was designed when the weight of train and rate of speed in fast passenger service had attained such proportions that greater boiler horse power was required for sustaining high speed than could be

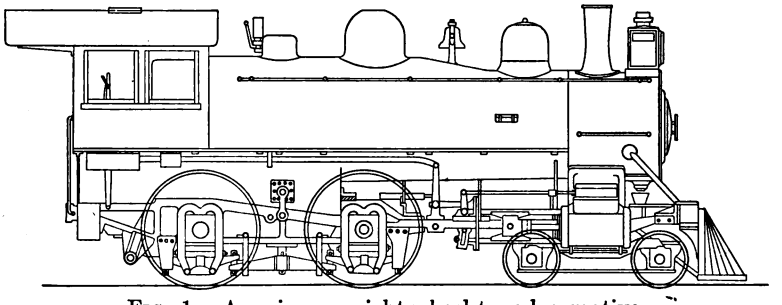


FIG. 1.—American or eight-wheel type locomotive.

obtained from a practicable design of a 4–4–0 type, but retaining all the advantages of a four-coupled engine. This type has been much in favor for high-speed passenger service.

3. Pacific Type (4–6–2), Fig. 5.—This is a development of the Atlantic type locomotive. It has a four-wheel front truck, six coupled drivers and a two-wheel trailing truck. It has the advantage of the Atlantic type in allowing the application of a large fire-box, while the required adhesion necessary for the larger hauling capacity is secured without placing too great a weight on any one pair of wheels. This type of locomotive is used extensively with heavy trains and for long hauls in modern high-speed passenger service.

4. Ten-Wheel Type (4–6–0).—This is a development of the American type. As in the case of the Pacific type, it is used

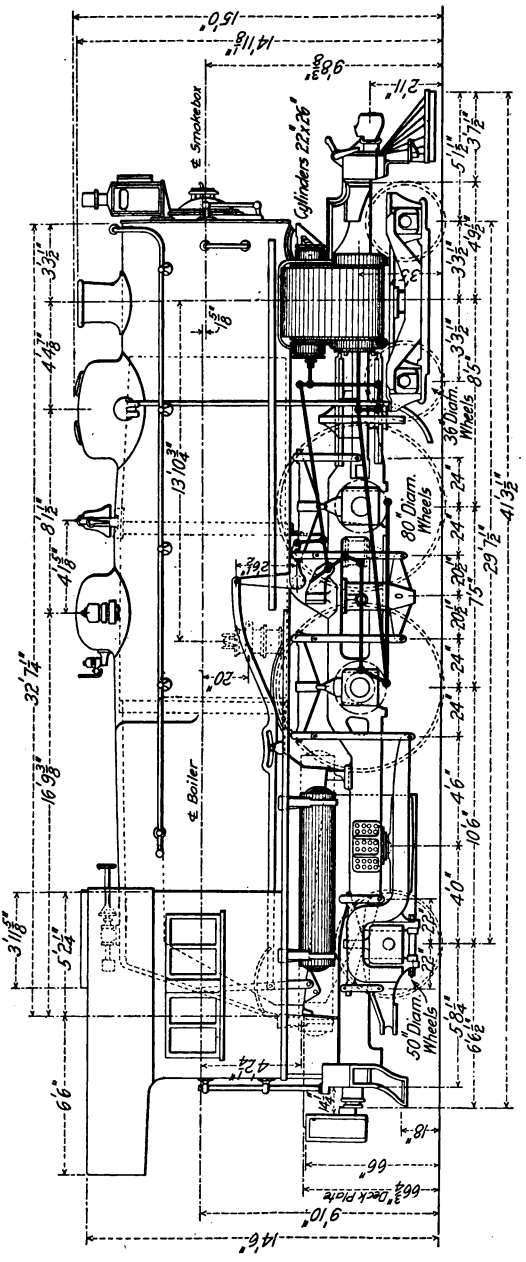


FIG. 2.—Modern Atlantic type locomotive.

in service where sufficient adhesion cannot be secured from the use of two pairs of drivers without overloading, but where sufficient boiler capacity can be secured with a fire-box placed between the frames.

5. Mogul Type (2–6–0).—This type of locomotive has a two-wheel front truck, six coupled drivers and no trailing truck. It is, to some extent, a modification of the ten-wheel type, in that it divides the weight necessary to secure adhesion over three pairs of drivers; but, unlike the previous type a greater proportion of the total weight of the engine is carried by the drivers, due to the use of a two-wheel front truck.

6. Consolidation Type (2–8–0), Fig. 3.—The Consolidation type was developed about 1875, when it became necessary to have

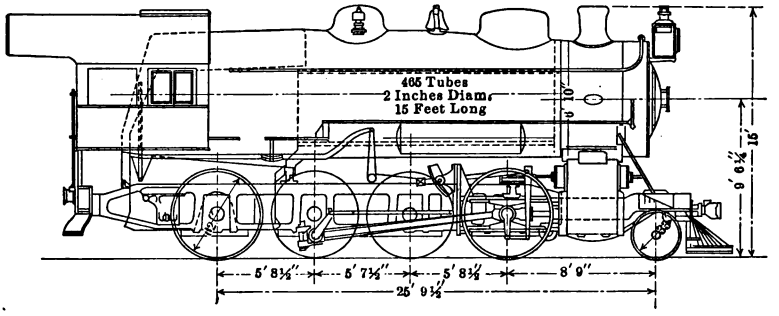


FIG. 3.—Consolidation type locomotive.

more powerful locomotives than the American, Ten-wheel or Mogul types. By the use of eight coupled driving wheels, it is possible to so divide the weight necessary to give adhesion for the power developed by the cylinders that the weight on any one wheel is not excessive. In some engines of this type it is impossible to give sufficient fire-box and boiler capacity to supply steam for the cylinders when working to their full capacity at any considerable speed. This is one of the serious limitations of this type. For many years it was the most important type of freight locomotive and to-day is in extensive use for handling slow-speed freight trains.

An example of what may be expected of a modern Consolidation locomotive is given in the following table compiled by The American Locomotive Co. from tests on the Kansas City Southern Railroad.

Average results	First district		Second district		Per cent. in favor new locomotives	
	New locomotives	Old locomotives	New locomotives	Old locomotives	First district	Second district
Average train loads, tons behind tender	2,369	1,709	1,224	992	38.6	23.4
Average ton-miles per lb. of coal	12.16	7.22	8.19	4.86	68.4	68.6
Average ton-miles per lb. of water	1.77	1.11	1.19	0.69	59.5	72.5

Total weight of new engines, 254,000 lb.; weight on drivers, 224,000 lb.; diameter of drivers, 56 in.; boiler pressure, 175 lb.; cylinders, 26 × 30 in.; maximum tractive power, 53,800 lb. The older locomotives have a total weight of 205,500 lb.; weight on drivers, 182,650 lb.; diameter of drivers, 55 in.; boiler pressure, 200 lb.; cylinders, 22 × 30 in.; maximum tractive power, 44,900 lb.

7. Mikado Type (2–8–2), Fig. 4.—The Mikado type was developed from the Consolidation. By the introduction of a two-wheel trailing truck, it is possible to use a larger fire-box than

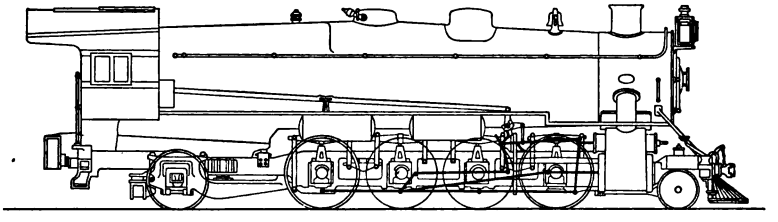
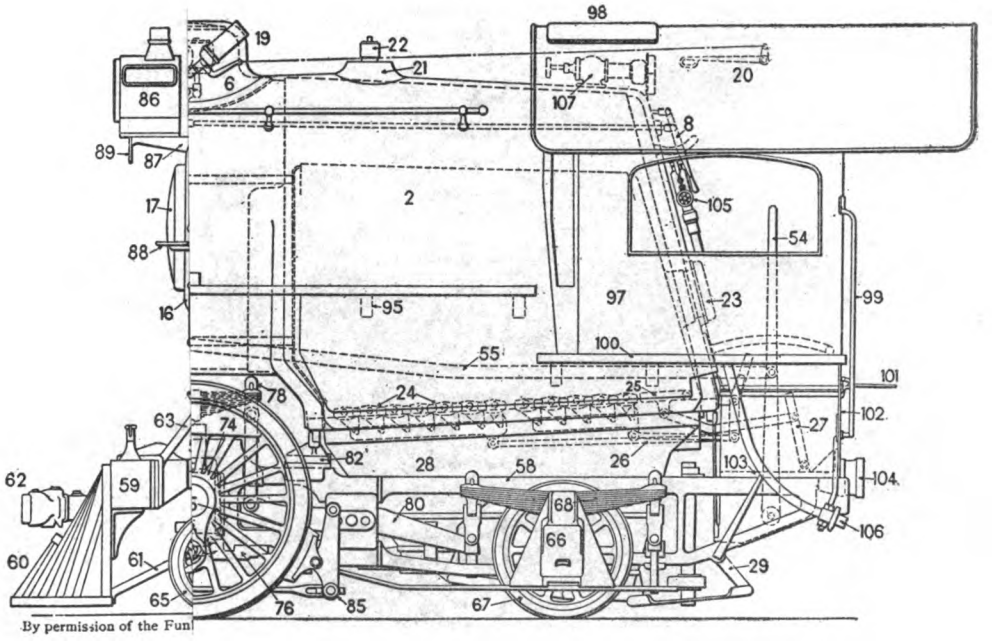


FIG. 4.—Mikado type locomotive.

in the Consolidation, thus increasing the working capacity of the locomotive. An increasing number of 2–8–2 locomotives are handling freight for long and heavy hauls. It is one of the most important types now in service.

Among the other types in restricted freight service may be noted the Mallet Articulated locomotives (Fig. 6), which are essentially two locomotives in one, there being two (or more) sets of cylinders and driving mechanism, each set depending on one common source of steam supply. The largest one built, which is now the heaviest locomotive in the world, weighs 853,000 lb., of which 761,600 lb. are on the driving wheels giving a total tractive force



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1 Boiler.
2 Firebox.
3 Fire tube.
4 Smoke box.
5 Smoke stack.
6 Dome.
7 Throttle chamber
8 Throttle lever.
9 Dry pipe.
10 Steam pipe.
11 Exhaust pipe.
12 Spark chute.
13 Spark cleaning ho
14 Diaphragm or plate.
15 Spark-arrester ne
Coupler.
Smoke box bumper brace
Front truck pedestal tie bar
Truck wheel.
Trailer truck oil box.
Trailer wheel.
Trailer truck spring.
Driving axle.
Driving wheel center.
Driving wheel counter-balance.
Driving wheel tire.
Crank pin.
Driving box.
Driving box shoes.
76 Frame pedestal brace.
77 Driving spring.
78 Driving spring hanger.
79 Driving spring saddle.
80 Driver equalizer.
81 Expansion plate.
82 Firebox expansion brace.
83 Air pumps.
84 Main reservoir.
85 Driver brakes.
86 Headlight.
87 Headlight bracket.
88 Step.
89 Number plate.
90 Bell.
91 Sandbox.
92 Sand pipes.
93 Step.
94 Running board.
95 Running board bracket.
96 Hand-rail.
97 Cab.
98 Cab ventilator.
99 Cab hand hold.
100 Cab floor.
101 Apron.
102 Cab bracket.
103 Deck plate.
104 Back chafing plate.
105 Injector.
106 Supply pipe.
107 Steam turret.

(*Facing page* 10.)

1 Boiler.
2 Firebox.
3 Fire tubes.
4 Smoke box.
5 Smoke stack.
6 Dome.
7 Throttle valve.
8 Throttle lever.
9 Dry pipe.
10 Steam pipe.
11 Main air pipe.
12 Sight feed.
13 Sight feed drip pipe.
14 Drifting valve or defector plate.
15 Superheater elements.
16 Superheater header.
17 Blower valve.
18 Blower pipe.
19 Cylinder cock.
20 Cylinder cock rod.
21 Cylinder cock lever.
22 Cylinder cock bracket.
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of 160,000 lb. Under certain traffic conditions the Mallets may lay claim to following advantages:

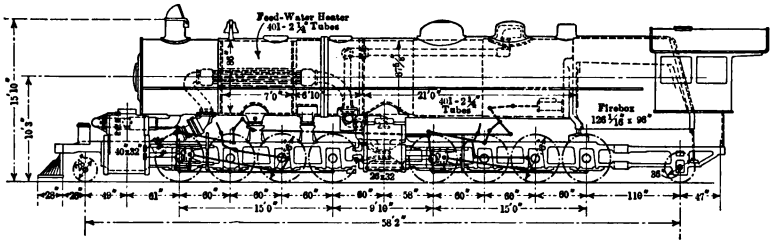
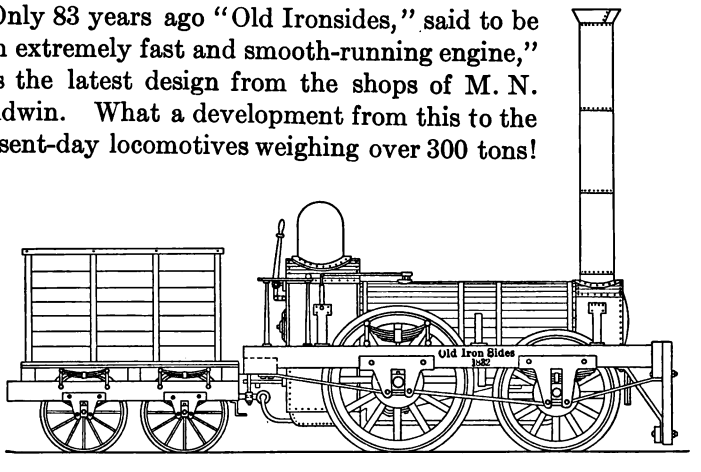


FIG. 6.—Mallet articulated locomotive. Type 2-8-8-2.

Reduced cost of operation by running fewer and heavier trains, increased capacity of a single-track division, reduction in overtime expense, and reduction in coal per ton-mile. However, the excessive weight and cost of repairs make their use out of the question on many roads.

Only 83 years ago "Old Ironsides," said to be "an extremely fast and smooth-running engine," was the latest design from the shops of M. N. Baldwin. What a development from this to the present-day locomotives weighing over 300 tons!



"Old Ironsides"—1832.

A careful study should be made of Fig. 5 with the accompanying names of parts. In Table II is listed the leading dimensions of certain modern locomotives, data from which is used in problem work in this text. Those given in columns (2) and (4) are superheated steam locomotives.

TABLE II.—DIMENSIONS OF CERTAIN MODERN LOCOMOTIVES

	(1) Pacific (K2) 4–6–2	(2) Pacific (K2sa) 4–6–2	(3) Atlantic (E2a) 4–4–2	(4) Mikado (A.L.Co.) 2–8–2
1. Total weight in working order, lb	272,000	293,200	184,167	315,000
2. Weight on drivers, working order, lb	178,500	179,900	110,000	243,000
3. Cylinders, in	24 × 26	24 × 26	20½ × 26	29 × 28
4. Diameter drivers, in	80	80	80	56
5. Heating surface in tubes (water side), sq. ft	4,420	3,453	2,471	3,740
6. Fire-box heating surface, sq. ft., including arch tubes	199	208	157	283
7. Heating surface of superheater, fire side, sq. ft	.....	989	.....	832
8. Total heating surface (based on water side of tubes), sq. ft.	4,619	4,650	2,628	4,855
9. Grate area total, sq. ft	55.4	53.7	55.5	66.7
10. Boiler pressure, lb. per sq. in.	205	205	205	170
11. No. of tubes	359	.....	315	238
12. No. of flues (for superheater)	.....	32	.....	40
13. Outside diameter of tubes, in.	2¼	2¼	2	2¼
14. Outside diameter of flues, in.	.....	5½	.....	5½
15. Length of tubes, in	251	250	180	228
16. Wt. of tender, loaded, lb	158,000	158,000	132,500	168,000

NOTE.—The weight of tender loaded does not, in any case, include stoker accessories, unless noted to that effect.

CHAPTER III

TRACTIVE EFFORT

FORCES WHICH MOVE A LOCOMOTIVE; FORCES ON PISTONS AND FRAMES; NEW GRAPHICAL METHOD FOR OBTAINING TURNING EFFORT FOR A COMPLETE REVOLUTION OF DRIVING WHEELS; RECENT METHODS FOR OBTAINING TRACTIVE EFFORT AT ANY SPEED; TABLES AND DIAGRAMS; COMPARISON OF DIFFERENT METHODS APPLIED TO AN ATLANTIC TYPE LOCOMOTIVE

Forces Moving the Locomotive.—We have mentioned the essential parts of a locomotive. The *mechanism* of a locomotive consists of (1) the crank, (2) connecting-rod,[1] (3) reciprocating parts (which include cross-head, piston-rod, and piston), and (4) frame (which includes the cylinder). Does the ordinary stationary reciprocating engine, being also a four-piece mechanism consisting of the above named parts, differ essentially from a locomotive? To the student familiar with the mechanics of machinery it will be apparent that, kinematically, it *does not* differ. The entire question reduces to one of relative motion. In the case of the stationary engine, the frame is the *fixed member;* while in the locomotive there is no fixed member, but the moving parts have the same relative motions as in the stationary engine. This will be apparent if we conceive the locomotive to be jacked up, so that the study of the parts may be independent of the rails. It is necessary to note one other condition. Just as the fly-wheel of a stationary engine may be considered an enlargement of the crank, so the driving wheels of a locomotive may be regarded as embodying the crank. Therefore, the four-piece mechanism of the one is equivalent to the four-piece mechanism of the other, and the principles involved in the running of a steam engine may be applied to the operation of a locomotive.

The question will now arise: If the mechanism of the one is similar to that of the other, why does the force driving the locomotive cause it to move along the rail, while the same force applied to a stationary engine turns the fly-wheel about a fixed

[1] Although the connecting-rod is here considered as a separate part, in making calculations for inertia, etc., some of its weight must be included with the reciprocating parts.

axis? An answer to this question will be evident by referring to Fig. 7. If the force on the piston is transmitted to a pair of driving wheels of the locomotive, these wheels when jacked up, will turn about their center, B; but if they rest upon the rails, the force still tends to turn the wheels about their axle as a fixed center, but this driving force is opposed by the resistance at the rails. The rail resistance is called the *adhesion* and equals the weight on the rail times the coefficient of friction at that point. If this resistance is sufficient to prevent slipping when the piston pressure is applied at the crank pin, then the center of the driving wheel instead of remaining fixed in respect to the earth (as it is when jacked up) will now move, due to the reaction of the force transmitted from the connecting-rod. Note, therefore, that the point of contact between wheel and rail is for the instant the center about which the wheel moves; and when it has moved, the next point of contact between wheel and rail becomes the instantaneous center.

This preliminary statement may lead to other questions, one of which may here be noted: Consider that an automobile (or a heavy wagon) is standing on a level roadway with power shut off but free to be moved. A person standing on the ground may move it by taking hold at or near the top of the rim of one of the wheels and pulling at right angles to a spoke at that point.

(*a*) Why can he not move it while applying the *same force* on the wheel, if he is standing or resting on the machine instead of on the ground? (Neglect the question of the additional weight caused by the man standing on the machine.)

(*b*) If standing on the ground, why can he not move the vehicle by pulling with equal force *below* the axle?

(*c*) If the pull *above* the axle is more effective than the pull *below* the axle as in *b*, above, why is not the force on the piston more effective in driving a locomotive when the crank pin is directly above the axle, than when directly below the axle? If the correct answer to (*c*) is not at once apparent, it is advised that the student return to this question after solving problem 1, page 16.

Analysis of Forces.—The crank pins on the same pair of driving wheels are 90° apart and when the locomotive is in motion under steam, the intensity of the force on the crank varies from instant to instant. The combined effect of the rotating forces should be less than the resisting forces required to slip the drivers on the

rails and under normal conditions, the adhesion of a wheel on a rail is from one-fourth to one-fifth of the weight of the wheel and its load.

If the piston on the right side is at the end of the stroke, the left crank will be vertically above or below the axle of the driving wheel as shown in Fig. 7, and the horizontal component of the force on the connecting-rod acts to produce rotation of the driving wheels about their axle. First, consider the ideal case where the force F on the piston is transmitted to the crank pin without frictional loss and the connecting rod is of infinite length.

The driving wheel acts as a lever AB with fulcrum at A; F' is the horizontal pressure, *induced by force F*, of the locomotive frame against the axle of the driving wheel. Taking moments about A, then for equilibrium:

$$F'r = F(r - a) \text{ or } F' = F - \left(\frac{a}{r}\right) F \qquad (1)$$

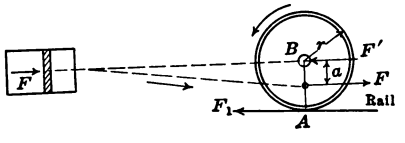


FIG. 7.—Forces to drive locomotive —left side.

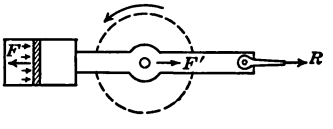


FIG. 8.—Horizontal forces on locomotive frame.

In the locomotive frame, Fig. 8, the force F' at the axle acts backward, but the force F acts in the opposite direction against the front cylinder head, giving for equilibrium of horizontal forces,

$$F - F' = R \text{ or } F' = F - R \qquad (2)$$

where R is the drawbar resistance or pull. Substituting F' from (1) in (2) gives,

$$F = \left(\frac{r}{a}\right) R \text{ or } R = F\left(\frac{a}{r}\right) \qquad (3)$$

or $Fa = Rr$, from which for equilibrium, the moment of the force F about the axle equals the moment of the pull R (at the axle) acting about the point of instant center at the track. Taking moments about B,

$$F_1 r = Fa \text{ or } F_1 = F\left(\frac{a}{r}\right) \qquad (4)$$

where F_1 is the force of adhesion. Combining (3) and (4), $F_1 = R$ proving directly that the drawbar pull is limited by rail friction. Therefore, the locomotive may move under the restricted conditions assumed, providing,

(*a*) Moment Fa is greater than moment Rr.
(*b*) Moment Fa is greater than moment $(F - F')r$.
(*c*) Moment Fa is less than moment F_1r.

Problem 1.—Draw a diagram for the connecting rod *above* the center line, on the right side of the locomotive and prove for this position that Eq. 3 holds true.

Problem 2.—Assume that the drawbar pull to start a train is 20,000 lb., diameter drivers 60 in., stroke 22 in., length of connecting rod 84 in. By aid of Eq. 3, find the diameter of cylinder, with steam pressure of 200 lb. per sq. in. which must be applied to start the train, assuming piston in 90° position, as in Fig. 7.

Forces for Any Crank Position.—The foregoing analysis assumes that the piston on one side is at the end of the stroke, the force turning the driving wheels being obtained from the other side of the locomotive. For any other position, steam will act on both pistons and the force from each will assist in producing rotation of driving wheels. Every crank angle position repre-

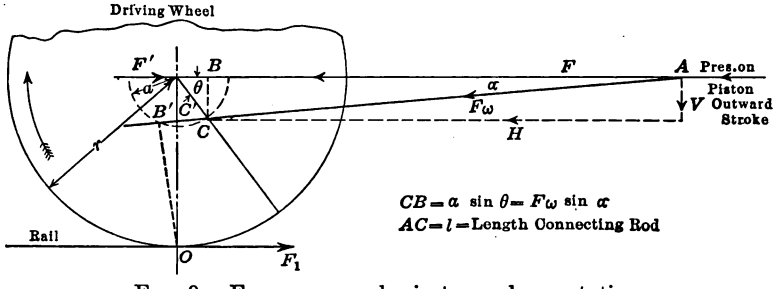


FIG. 9.—Forces on crank pin to produce rotation.

sents a different turning effort and therefore a different tractive effort exerted at the point of contact between driving wheels and rails.

Steam in the head end of the cylinder produces two equal and opposite pressures, the one on the forward head tending to move the locomotive forward and the other on the piston tending to move the locomotive backward, this latter force being transmitted through piston rod, crosshead and connecting-rod to the crank pin. There is a vertical component of the force on the driving wheels, but this force does not act to move the engine forward or backward.

Referring to Fig. 9: The piston pressure F acting along the connecting rod and represented as AC (or Fw) may be resolved

into a horizontal component H and a vertical component V. Let the force of adhesion at the point O of rail contact be represented by F_1, the radius of driver r, radius of crank-pin circle a, and crank angle θ. Then if $F_1 r$ equals or is greater than H ($a \sin \theta$), no force will be transmitted along the side rod to the next pair of drivers; but, if $F_1 r$ (the moment which indicates the tendency to slip the drivers) is less than the moment $H \times a \sin \theta$, a fractional part x of H will pass through the parallel or side rod to the crank pin of the second set of drivers, and if the adhesion on this second set is not sufficient to prevent the tendency to slip caused by xH, then a part y of xH will pass to a third set of drivers, and so on.

The distribution of horizontal crank-pin pressures will be as follows:

First set drivers	$(1 - x)H$.
Second set drivers	$(1 - y)xH$.
Third set drivers	xyH.

The force Fw in starting will tend to produce motion about the point O and its moment will be $F_w \times OB'$, OB' being perpendicular from O to the connecting-rod produced and O acting as an instantaneous center. From the similar triangles ABC and $OB'C'$

$$AB : OB' = AC : OC'; \text{ or } AB \times OC' = OB' \times AC \quad (5)$$

But

$$AB = F \text{ and } AC = F_w; \text{ or moment } F_w \times OB' = F \times OC' \quad (6)$$

Therefore, the moment of the force along the connecting-rod acting about O equals the force (F) on the piston times the distance (OC') from O to the intersection of the connecting-rod extended to meet the vertical line through the center of the driving wheel.

By changing the algebraic sign of F, Eq. 6 applies to points on the inward stroke when the crank pin is above the line of centers and the pressure of the axle box against the frame is opposite in direction to that shown in Fig. 9.

TRACTIVE FORCE FOR A COMPLETE REVOLUTION—GRAPHICAL SOLUTION

Neglecting friction (except that at the track) and the effect of inertia of moving parts, all points for the tractive force during a complete revolution of the drivers may be determined graphically. Fig. 10 shows to scale part of a driving wheel of a loco-

motive, with the crank-pin path divided into thirty-two equal parts. To the right at **B** on the center line of the cylinder are laid off the corresponding piston positions, and directly above at **A** are shown indicator cards from a locomotive with cylinders 24 × 26 in., the cards being taken at 80 r.p.m. (16.6 m.p.h. and at approximately 30 per cent. cut-off.

By projecting the piston positions on the card, ordinates are scaled off representing the *effective* driving pressure at each crank-pin position. This pressure will be negative for positions in which the compression pressure on one side of the piston is greater than the steam pressure on the other side, as in position of ordinate 13.

Take, for example, position 4 of the crank pin: First, draw PM at any convenient point on the line of the rail and perpendicular to PA'. Produce the center line of the connecting-rod until it intersects $A'H$ at 4 and project $A'4$ on PM, thus making $PE = A'4$. Lay off MV equal to ordinate 4 of the indicator card and draw the line $E'V$ through V parallel to PM. Draw PL through V and erect a perpendicular to PM at E intersecting PL at W. Then:

1. $EE' = MV$ = effective cylinder pressure.
2. EW = axle box pressure = F', Figs. 7 and 9.

Proof.—In the similar triangles.

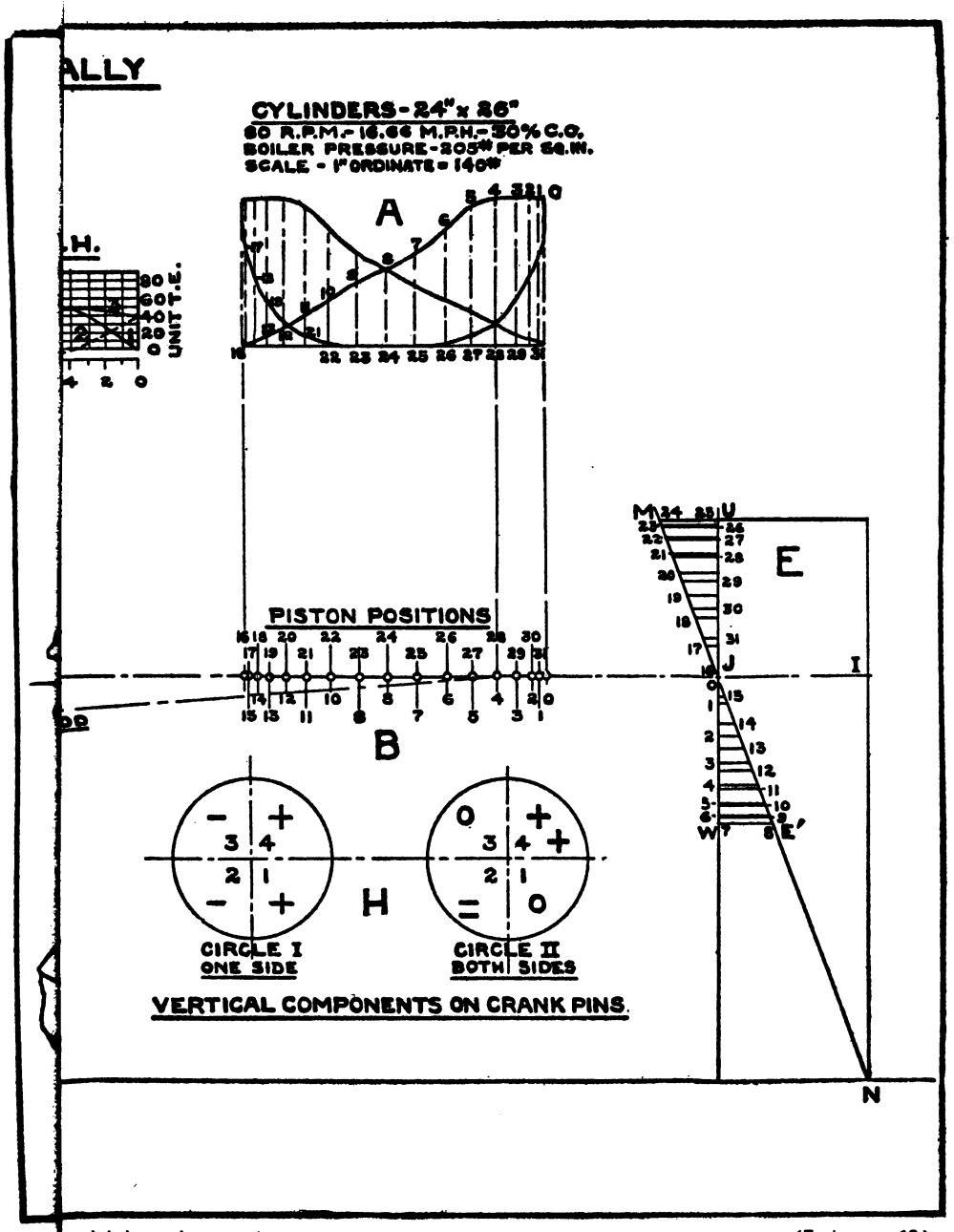
$$PEW \text{ and } PMV,\ MV:EW::MP:PE \text{ or } F:EW=r:PE$$

or

$$EW = \frac{F \times PE}{r}$$

From Eq. 1, $F' = \dfrac{F(r-a)}{r}$ which applies to the position of 90° crank. $F(r-a)$ is the moment about the point of contact of wheel and rail. It has been shown in Fig. 9 that for any other position this equals the force F times the distance from the rail to the intersection of the connecting-rod extended until it meets the vertical line through the center of axle (or $F \times PE$). Hence $EW = F'$ = axle-box pressure.

3. Since $F' = F - R$ or $R = F - F'$ (Eq. 2), $WE' = R = EE' - EW$. Forces in the above are in pounds per square inch of piston area which are the same units as the scale of pressure on the indicator card. If desired, total piston pressure may be used to its proper scale but this would alter neither the method nor the shape of the turning effort curve.



n driving a locomotive.

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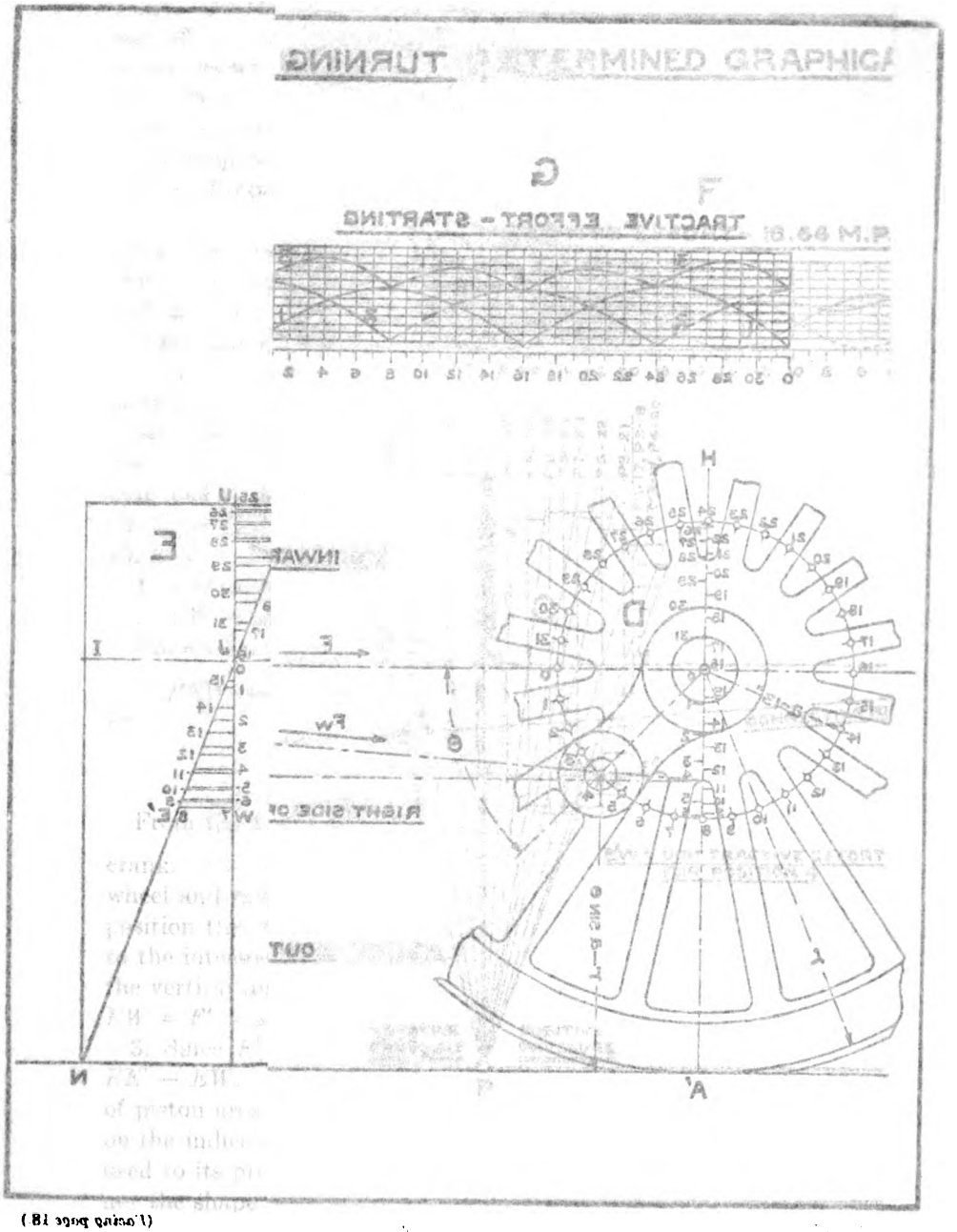

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The effort for other positions is found in the same way. After passing position 12 on the outward stroke, the driving pressures become negative, because the force on the compression side of the piston is greater than the force on the other side. These net effective forces are laid off to the left of PM. Efforts for the inward stroke are shown above the line of centers and since the two indicator cards are practically identical, the values for crank effort are the same numerically as for the outward stroke.

At **F**, Fig. 10, are shown two curves: (1) represents the variation in effort due to one cylinder of the locomotive, and is found by erecting on each of the thirty-two crank-pin positions the corresponding effort as found on the diagram **C**. Curve (2) shows the turning effort for the opposite side of the locomotive; (3) represents the variation of the combined effort of the two cylinders, and is found by adding the ordinates of any point in curves (1) and (2) to obtain the corresponding ordinate on curve (3).

At **E**, Fig. 10, is shown the construction for obtaining the effort for a full-stroke card, IJ representing the constant effective pressure throughout the stroke. Drawing UW and MN as shown, project the intercepts from $A'H$ (in **D**) as was done for point 4 in **C** (giving in that case a distance $PE = A'4$), and find the lengths of the projecting lines contained between MN and UW in **E**. These lengths represent the tractive efforts for a single cylinder and are plotted in curve (1) in diagram **G**. Curve (3) is for both sides and may be drawn from the addition of the zero and eighth, first, and ninth, etc., ordinates of the single-cylinder curve (1), or by adding ordinates for any point in curves (1) and (2), as explained above. Curve (3) has a maximum at or near the fourth, twelfth, twentieth, and twenty-eighth crank-pin positions. Were it not for the angularity of the rod, curve (3) would be of the same height at these four points; but, as is seen, this element has considerable influence on the height of the ordinates at these positions of the crank pin.

Any force applied to the driving wheel in such a manner as to assist rotation in the direction the wheel is turning, will increase the traction and a force which resists rotation will decrease traction. Beginning with the *following* crank pin in the first quadrant on the outward stroke, we find the vertical component assisting rotation; in the second and third quadrants, resisting rotation and in the fourth assisting rotation, shown in the four quadrants of the circle 1 in **H**, Fig. 10, for a single cylinder.

The combined effect of the vertical components on both sides is found by adding the effects in consecutive quadrants, beginning with the first. Thus the combined effect, when the following crank pin was in the first quadrant, would be zero when the vertical components of first and second quadrants neutralized each other and throughout that quadrant the sum of the combined components will be zero. This explains why curve (3) in first quadrant (from 0 to 8 in **G**) becomes depressed in the second quadrant (from 8 to 16) due to the fact that both are negative in the second quadrant. In the third, the effect on the two sides is to neutralize one another and in the fourth quadrant, when both vertical components are positive, the greatest turning effort occurs and consequently the greatest tendency for the driving wheels to slip.

Observe that the stresses on one side vary through a wide range from a maximum to a minimum. This variation has been in some cases sufficient to account for many serious and repeated breakages in reciprocating parts and in crank pins.

Problem 3.—With data from Table II, Chap. II, for an E2a locomotive, draw a turning effort curve *in starting*, following the method shown in Fig. 10. Take drop in steam pressure between boiler and cylinder as 10 lb. and length of connecting rod 110 in.

Inertia Effect.—At higher speeds, the inertia effect of the reciprocating parts, when combined with the curves plotted in **F** (Fig. 10), influences the shape of curve 3, tending to smooth out the variations in torque effect. At a speed in miles per hour equal to the driving wheel in inches (called "diameter speed") the horizontal inertia force, P_1 etc., at the end of the stroke may be found as follows:

	Front end of stroke	Back end of stroke
For unbalanced revolving weight,[1]	$P_1 = 3.2Wr$	$P_1 = 3.2Wr$
For reciprocating parts,	$P_2 = 3.2Wr\left(1+\frac{r}{l}\right)$	$P_2 = 3.2Wr\left(1-\frac{r}{l}\right)$
For connecting rod	$P_3 = 3.2Wr\left(1+\frac{Xr}{l_2}\right)$	$P_3 = 3.2Wr\left(1-\frac{Xr}{l_2}\right)$

Where: r = radius of crank; l = length of connecting-rod; X = distance of crank to center of gravity of connecting-rod; all in inches. W = weight producing inertia effect.

The Vertical Component.—Referring to Fig. 9, the vertical pressure V (or F_v) reacting against the guides is equal numerically to the vertical component of the force acting on the crank pin, the latter force tending to add to or subtract from the effective

[1] Derived in Eq. 98*c*, page 239.

weight on the track. In running forward, this force acts down, increasing the vertical pressure on the rail and when running backward it acts up, decreasing that pressure.

The vertical pressure on the guide and its corresponding equal reaction at the rail may be derived as follows:

$$F_v = V = F \tan \alpha, \text{ but } l \sin \alpha = a \sin \theta \text{ or } \sin \alpha = \frac{a}{l} \sin \theta$$

$$\cos \alpha = \sqrt{1 - \frac{a^2}{l^2} \sin^2 \theta}$$

$$\text{since } \tan \alpha = \frac{\sin \alpha}{\cos \alpha}$$

$$V = F \frac{a}{l} \times \frac{\sin \theta}{\sqrt{1 - \frac{a^2}{l^2} \sin^2 \theta}} \qquad (7)$$

Except in starting and at slow speeds, the steam pressure F is changing during expansion but in starting it is nearly constant throughout the stroke, the vertical component V being a maximum when $\theta = 90°$.

Therefore, $V = F \frac{a}{l} \times \frac{1}{\sqrt{1 - \frac{a^2}{l^2}}}$; since $\frac{a^2}{l^2}$ is very small, we may write,

$$V = F \frac{a}{l} \qquad (8)$$

for the crank at the 90° point at either the forward or backward stroke. At dead centers, $\sin \theta = 0$ and $V = 0$. This pressure increases to a maximum at 90°, the pressure being on the top guide running forward and the lower guide running backward.

Equation 8 gives the total vertical component pressure in starting. As the center of the guide is about one-fourth the distance from the truck center to the center of the equalized weight, we may consider that it will take three-fourths of this thrust off the truck and one-fourth off the drivers, which will give sufficiently close results. Equation 8 for any angle θ would, therefore, be multiplied by $\frac{3}{4}$

$$F_v = \tfrac{3}{4} F \frac{a}{l} \sin \theta \qquad (8a)$$

Slipping.—As pointed out in Chap. I, the adhesion fixes the limit of the cylinder power and hence the allowable turning effort for any locomotive. It is common to use a factor of adhesion of from 4.5 to 5.0 or even over, but locomotives are in success-

ful operation using a factor of 4.0 to 4.3. Remembering that this factor represents the ratio of weight on the driving wheels to the resistance at the rails before slipping actually occurs, it will be apparent that the difference between the lowest possible factor representing slipping conditions and the factor used in any particular design may be considered as the *margin for safe design.* This margin may be carried to extremes at the sacrifice of unproductive weight and cost.

Opposed to the rotating forces, which we have found to vary widely throughout a complete revolution, are the following factors which determine the adhesive weight and therefore the resistance to slipping at any instant. The first two are independent of speed and the third is effective only at high speed.

1. The static or dead load on drivers.

2. The vertical thrust on the crank pin, using Eq. 8*a*, remembering it is always positive when running ahead and negative when running back.

3. The vertical effect at "diameter speed," of the excess balance, equals 1.6 × stroke × excess balance × sin θ. The excess balance = (approx.) Weight connecting rod + weight reciprocating parts − vertical balance of rod − $\frac{\text{total wt. of engine}}{400}$.

The algebraic sum of 1, 2 and 3 for any position, times the coefficient of rail friction gives the resistance to slipping and this product should be greater than the rotative effort for any crank position. Data for weight of connecting rod and reciprocating parts for five modern locomotives are given in Table XXIV, page 242, and a discussion of excess balance weight will be found in Chap. XIV.

TRACTIVE EFFORT IN STARTING AND AT LOW SPEEDS[1]

In starting and at low speeds the reverse lever is pushed far forward "down in the corner," so that steam is admitted to the cylinders at, or nearly at, boiler pressure and this pressure continues to force the piston through nearly the entire stroke. However, instead of obtaining at the track 100 per cent. effective boiler pressure there are losses in transmission of the steam to the cylinders, drop in pressure near the end of the stroke and internal friction of the moving parts which combine to reduce the effec-

[1] The author has avoided the term "tractive power," so commonly used. The terms "tractive effort" and "tractive force" are used in this text in preference to other terms.

tiveness of the steam at boiler pressure by some 15 to 20 per cent. To obtain the rated tractive force, the theoretical tractive force is therefore multiplied by a constant, usually 0.85, but 0.80 may be used as a more conservative value.

The work done in starting by the two cylinders in one revolution of the drivers, equals the total uniform pressure on the pistons times the distance passed over by the pistons in one revolution of the drivers $= \frac{4P\pi d^2 s}{4}$ and this equals the pull or theoretical tractive force (*T.T.F.*) acting through one revolution.

$$T.T.F.\ \pi D = P\pi d^2 s$$

$$T.T.F. = \frac{Pd^2 s}{D} \tag{9}$$

or the rated tractive force,

$$T.F. = \frac{0.85Pd^2 s}{D} \tag{10}$$

Where

P = boiler pressure, lb. per sq. in.
D = diameter of drivers, in.
d = diameter of cylinders, in.
s = stroke of piston, in.

Tractive Effort Decreases as Speed Increases.—Equation 10 expresses the condition at slow speeds; that is, up to 50 or 60 r.p.m. As the speed increases, the available supply of steam is used more rapidly until the boiler can no longer generate the amount necessary for the cylinder at full stroke.[1] The cut-off must, therefore, be shortened as the speed increases and this shorter cut-off gives less work per stroke in the cylinders and the tractive force is correspondingly decreased. Equation 10 must therefore be multiplied by a speed factor, the value of which may be determined on different assumptions, but based, in each case, on the work done per piston stroke. We may write

$$T.F. = \frac{0.85Pd^2 s}{D} \times \textit{speed factor} \tag{11}$$

The Baldwin Locomotive Works' Factors.—As pointed out, the available tractive force falls off slowly as the speed increases, until a point is reached at which the boiler can no longer supply the steam required by the cylinders at full stroke. To attain higher speeds the cut-off must be shortened, after which the available

[1] For method of determining this speed, see Eq. 97, page 222.

tractive force falls more rapidly. Under these conditions, the tractive force is dependent not only on the cylinder and driving wheel dimensions (see Eq. 10) but also on the steaming capacity of the boiler. As shown by the curves, Fig. 11, the available tractive force at any speed will depend on the relation between the rated tractive force and the total heating surface and each curve corresponds to a different value of this relation. The vertical scale measures the available tractive force as a percentage of the rated tractive force, while on the horizontal scale the speed is measured in miles per hour. *The curves assume that at the high*

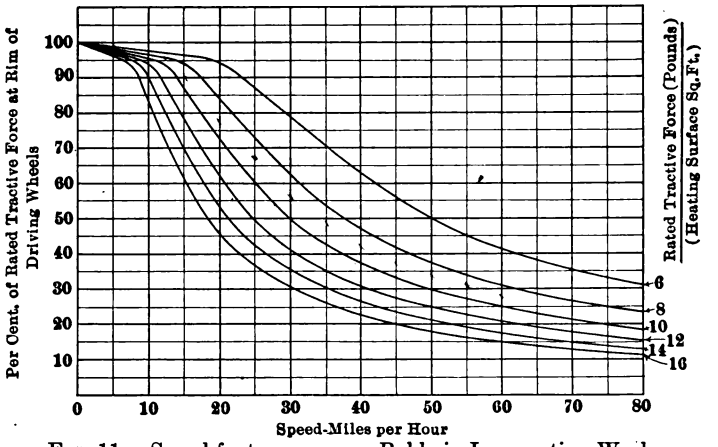


FIG. 11.—Speed factor curves.—Baldwin Locomotive Works.

speeds 1 *hp. can be developed at the tread of the driving wheels for every* 2½ *sq. ft. of heating surface, and they allow for a lower efficiency at slow speeds.*

In assuming that the steaming capacity is directly proportional to the total heating surface, it is essential that the ratio of grate area to heating surface be properly suited to the quality of the fuel. It is also assumed that sufficient fuel can be fired to enable the steam production to be pushed to the limit set by the heating surface. The average maximum evaporation in a coal-burning locomotive may be taken as 12 lb. of water per square foot of heating surface per hour, and 18 lb. when burning oil.

Suppose it is desired to find the available tractive force at a speed of 40 m.p.h. for a locomotive using saturated steam, having the following dimensions:

Cylinders, 22 × 28 in.	Steam pressure, 200 lb.
Driving wheels, 69 in. diam.	Heating surface, 4150 sq. ft.

By Eq. 10 (or from Table III) the rated tractive force of this locomotive is 33,400 lb. The ratio of rated tractive force to heating surface is therefore $\frac{33400}{4150} = 8.0$. Referring to the curve, Fig. 11, it is seen that the vertical line representing 40 m.p.h. intersects the curve marked 8, on a horizontal line representing 47 per cent. Hence, the tractive force developed by this locomotive, at a speed of 40 m.p.h., will be $33,400 \times 0.47 = 15,700$ lb.

In order that a locomotive may employ all of its rated tractive force in hauling a train, not more than about 25 per cent. of the adhesive weight can be utilized as tractive force.

In calculating the tractive force developed by a *superheater locomotive* at comparatively high speeds, account must be taken of the increased capacity due to the use of superheated steam. Given two locomotives of the same general dimensions, one equipped with a superheater and one without, the superheater locomotive will have a smaller amount of water-evaporating surface than the other, because part of its water-evaporating surface is replaced by superheating surface. With a Schmidt fire-tube superheater the superheating surface provided is usually about 20 to 22 per cent. of the water-evaporating surface.

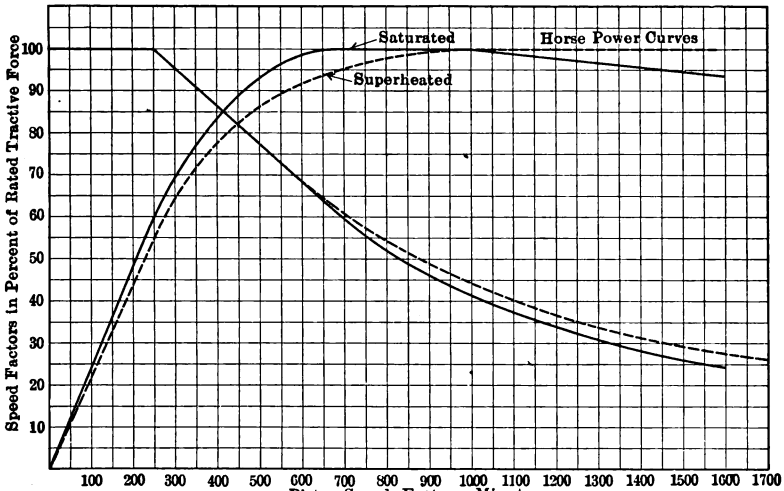
In selecting the proper curve from Fig. 11, for a superheater locomotive, the rated tractive force must be divided by what has been termed the "equivalent heating surface." This is calculated by multiplying the water-evaporating surface by 1.60 for about 200° superheat and 200 lb. steam pressure. The increased capacity of the superheated over the saturated steam locomotive is from 25 to 30 per cent. or more, as will be pointed out in Chap. XI.

The maximum power developed by a compound locomotive using saturated steam may be taken as approximately 20 per cent. greater than that of a single expansion locomotive of the same general dimensions. Therefore to obtain the "equivalent heating surface" of a compound locomotive using saturated steam, the actual heating surface is multiplied by 1.20. The product is then divided into the rated tractive force, in order to select a suitable curve from the diagram.

The American Locomotive Company uses speed factors derived from a large number of tests made under widely varying conditions and these factors for saturated and superheated steam are shown in Fig. 12. It may be noted that the curves shown in this chapter do not take account of the resistance of loco-

motive and tender, the effort in each case being at the rim of the driving wheel.[1]

The Kiesel Tractive Force Formula.[2]—In the following, it is assumed that a locomotive evaporates an approximately uniform amount of water per hour at speeds above 20 m.p.h. For prolonged runs the actual water evaporated may be taken as 10 lb. per sq. ft. of heating surface per hour. For runs of about 2 hr. duration about 20 per cent. more can be realized and for runs of 1 hr. or less even 50 per cent. more can be obtained.



Piston Speed, Feet per Minute																
100	200	300	400	500	600	700	800	900	1000	1100	1200	1300	1400	1500	1600	1700
% H.P. Superheated																
.225	.449	.643	.776	.868	.920	.952	.975	.993	1.00	1.00	1.00	1.00	1.00	1.00	1.00	1.00
% H.P. Saturated																
.245	.490	.691	.837	.935	.987	1.00	1.00	1.00	1.00	.99	.978	.968	.957	.947	.935	
Speed Factor-Superheated																
1.00	1.00	.954	.863	.772	.682	.606	.542	.490	.445	.405	.371	.342	.318	.297	.278	.262
Speed Factor-Saturated																
1.00	1.00	.954	.863	.772	.630	.590	.517	.460	.412	.372	.337	.307	.283	.261	.241	

FIG. 12.—Speed factor and horse-power curves—American Locomotive Co.

[1] A graphical solution of Eq. 11, using the A. L. Co. factors, prepared by Mr. L. R. Pomeroy, appears in Railway Age Gazette, Mech. Edition, Sept., 1915.

[2] Developed by Mr. W. F. Kiesel, Jr., Asst. Mech. Engr., Pennsylvania Railroad. Equation 12 is deduced from Eq. 9, expressed in terms of total boiler steaming capacity, cylinder power, and the speed. It does not apply to speeds less than 15 to 20 m.p.h., the exact speed being determined by calculating the speed of the locomotive at which the boiler can supply steam at full stroke without drop in pressure. (See page 222.) In plotting, round off the upper part of the Kiesel curve bringing it tangent to a line plotted from Eq 10, for the rated tractive force.

Let: $T.F.$ = cylinder tractive force in lb.
D = driver diam. in in.
d = cylinder diam. in in.
l = cylinder stroke in in.
H = total boiler heating surface in sq. ft.[1]
P = initial cylinder pressure in lb. (considered as 10 lb. less than boiler pressure).
K = evaporation per hour in lb. per sq. ft. of external heating surface.
w = weight of 1 cu. ft. of steam in lb. (Steam Tables.)
V = speed in miles per hour.
$M = \frac{d^2 l}{D}$.

The formula for cylinder tractive force deduced for average conditions is,

$$T.F. = \frac{2PM}{1 + \frac{110w}{3} \times \frac{MV}{KH}} \qquad (12)$$

Based on an evaporation of 10 lb. of water per square foot of heating surface per hour, the formula for tractive force of locomotives which may be used regularly or in emergency cases in passenger service may be expressed from Eq. 12, as follows:

Penn. R.R. class locomotive	Tractive force
D16d	$\frac{38930}{1 + 0.0987V}$
E2a	$\frac{53305}{1 + 0.0873V}$
E3a	$\frac{61347}{1 + 0.1005V}$
E6s (based on 22 × 26 in. cyl. and 3348 sq. ft. H.S.)	$\frac{61386}{1 + 0.0672V}$
E6s (Based on 23½ × 26 in. cyl. and 3577 sq. ft. heating surface)	$\frac{70000}{1 + 0.0715V}$
F3c	$\frac{70450}{1 + 0.1234V}$
G4	$\frac{73406}{1 + 0.15V}$
K2	$\frac{73008}{1 + 0.09022V}$
H6a	$\frac{94380}{1 + 0.1435V}$
H8b	$\frac{101440}{1 + 0.1253V}$

[1] In case of superheated steam locomotive, multiply the superheated surface by 1.5 and add this to the other heating surfaces. For tubes, take external heating surface.

TABLE III.—Rated Tractive Force of Locomotives (Eq. 10, page 23)

Boiler Pressure 100 Lbs. per Sq. In.

Cylinder		Diameter of driving wheels—inches																					
D. in.	S. in.	44	48	50	52	54	55	56	57	58	59	60	62	64	66	68	70	72	74	76	78	80	82	84
14	**24**	**9100**	**8350**	**8000**	**7700**	**7400**	**7250**	**7150**	**7000**	**6900**	**6800**	**6650**	**6450**	**6250**	**6050**	**5900**	**5700**	**5550**	**5400**	**5250**	**5150**	.....	.....	.....
15	24	10450	9600	9200	8850	8500	8350	8200	8050	7900	7800	7650	7400	7200	6950	6750	6550	6400	6200	6050	5900	.....	.....	.....
16	24	11850	10900	10450	10050	9650	9500	9350	9200	9000	8850	8700	8400	8150	7900	7700	7450	7250	7050	6850	6700	.....	.....	.....
17	24	13400	12300	11800	11350	10900	10700	10550	10350	10150	10000	9850	9500	9200	8950	8650	8400	8200	7950	7750	7550	.....	.....	.....
18	24	15000	13800	13200	12700	12250	12000	11800	11600	11400	11200	11000	10650	10350	10000	9750	9500	9200	8950	8750	8500	.....	.....	.....
20	24	18500	16950	16250	15700	15100	14800	14550	14300	14050	13850	13600	13150	12750	12400	12000	11650	11350	11050	10750	10500	.....	.....	.....
22	24	22300	20500	19650	18900	18200	17900	17600	17300	16950	16700	16400	15900	15400	14950	14500	14100	13700	13350	13000	12650	.....	.....	.....
23	24	24500	22500	21500	20700	19900	19600	19200	18900	18600	18300	18000	17400	16800	16350	15850	15400	15000	14600	14200	13800	.....	.....	.....
18	**26**	.....	.....	**14350**	**13800**	**13300**	**13050**	**12800**	**12550**	**12350**	**12100**	**11900**	**11550**	**11200**	**10850**	**10500**	**10250**	**9950**	**9700**	**9450**	**9200**	**8950**	**8750**	**8550**
19	26	.....	.....	16000	15400	14850	14550	14250	14000	13750	13500	13300	12850	12500	12100	11700	11400	11100	10800	10500	10250	10000	9750	9550
20	26	.....	.....	17700	17100	16400	16100	15800	15500	15300	15000	14750	14250	13800	13400	13000	12650	12300	11950	11650	11350	11050	10800	10500
21	26	.....	.....	19500	18800	18100	17750	17400	17100	16800	16500	16250	15700	15250	14750	14350	13900	13550	13150	12850	12500	12200	11900	11600
22	26	.....	.....	21300	20600	19800	19500	19100	18700	18400	18100	17800	17250	16700	16200	15800	15300	14900	14500	14100	13700	13400	13100	12750
23	26	.....	.....	23300	22600	21700	21300	20800	20500	20100	19800	19500	18900	18250	17700	17200	16700	16200	15800	15400	15000	14600	14250	13900
24	26	.....	.....	25500	24600	23700	23300	22800	22300	21900	21600	21200	20600	19900	19300	18700	18200	17700	17200	16750	16300	15900	15500	15200
25	26	.....	.....	27600	26600	25600	25100	24700	24300	23800	23400	23000	22300	21700	21000	20300	19800	19250	18700	18200	18700	17300	16900	16500

TABLE III.—Rated Tractive Force of Locomotives.—(*Continued*)

Boiler Pressure 100 Lbs. per Sq. In.

Cylinder		Diameter of driving wheels—inches																				
D. in.	S. in.	56	57	58	59	60	61	62	63	64	65	66	67	68	70	72	74	76	78	80	82	84	86
17	**28**	**12300**	**12050**	**11850**	**11650**	**11450**	**11300**	**11100**	**10900**	**10750**	**10600**	**10400**	**10250**	**10100**	**9850**	**9550**	**9300**	**9050**	**8800**	**8600**	**8400**	**8200**	**8000**
18	28	13750	13550	13300	13100	12900	12700	12500	12300	12100	11900	11700	11500	11350	11000	10700	10400	10150	9900	9700	9450	9200	9000
19	28	15400	15100	14850	14600	14300	14100	13900	13700	13450	13200	13000	12850	12700	12300	11950	11650	11300	11000	10750	10500	10250	10000
20	28	17000	16700	16400	16100	15900	15600	15350	15100	14900	14650	14450	14200	14000	13600	13200	12850	12550	12200	11900	11600	11350	11050
21	28	18750	18400	18100	17800	17500	17200	17000	16700	16400	16150	15900	15700	15500	15000	14600	14200	13800	13450	13100	12800	12500	12200
22	28	20500	20200	19800	19500	19100	18800	18500	18300	18100	17800	17500	17250	17000	16500	16100	15600	15200	14800	14500	14100	13750	13500
23	28	22500	22100	21700	21300	21000	20600	20300	20000	19700	19400	19100	18800	18500	18000	17500	17000	16600	16100	15800	15350	15000	14700
24	28	24500	24100	23600	23300	22900	22500	22100	21800	21400	21100	20800	20500	20200	19600	19100	18500	18000	17600	17100	16700	16350	16000
26	28	28800	28300	27800	27300	26800	26400	26000	25600	25200	24800	24400	24100	23700	23000	22400	21800	21200	20700	20100	19700	19200	18700
28	28	33400	32800	32200	31700	31200	30700	30200	29700	29200	28800	28300	27900	27500	26700	26000	25300	24600	24000	23400	22800	22200	21800
18	**30**	**14750**	**14500**	**14250**	**14000**	**13800**	**13550**	**13350**	**13100**	**12900**	**12700**	**12500**	**12300**	**12150**	**11800**	**11500**	**11200**	**10900**	**10600**	**10350**	**10100**	**9900**	**9650**
20	30	18250	17900	17600	17300	17050	16800	16500	16200	15950	15750	15500	15300	15050	14600	14200	13850	13500	13100	22800	12500	12200	11900
22	30	22000	21600	21300	20900	20600	20200	19900	19600	19300	19000	18700	18400	18150	17650	17200	16700	16250	15850	15400	15050	14700	14350
24	30	26200	25800	25300	24900	24500	24100	23700	23300	23000	22600	22200	21900	21600	21000	20400	19850	19350	18800	18350	17900	17500	17100
25	30	28500	28000	27500	27000	26600	26200	25700	25300	24900	24500	24200	23800	23500	22700	22100	21500	21000	20500	19950	19450	19000	18500
26	30	30700	30200	29600	29200	28700	28200	27700	27400	27000	26600	26100	25700	25300	24600	23900	23300	22700	22100	21500	21000	20500	20050
27	30	33200	32600	32000	31500	31000	30500	30000	29500	29000	28600	28200	27700	27300	26600	25800	25100	24500	23800	23200	22600	22100	21600
28	30	35700	35100	34500	33900	33400	32800	32300	31800	31200	30800	30300	29800	29400	28600	27800	27100	26300	25600	25000	24400	23800	23200
19	**32**	**17550**	**17250**	**16950**	**16650**	**16350**	**16100**	**15850**	**15600**	**15350**	**15100**	**14900**	**14650**	**14450**	**14000**	**13650**	**13300**	**12900**	**12600**	**12300**	**12000**	**11700**	**11400**
20	32	19400	19100	18750	18450	18150	17850	17550	17300	17000	16700	16500	16250	16000	15600	15150	14700	14350	13950	13600	13250	12950	12650
21	32	21400	21000	20700	20300	20000	19700	19400	19050	18750	18450	18200	17900	17650	17150	16650	16200	15800	15400	15000	14650	14300	13950
22	32	23500	23100	22700	22300	22000	21600	21200	20900	20600	20200	19950	19650	19350	18800	18300	17800	17300	16900	16500	16050	15650	15300
23	32	25700	25300	24800	24400	24000	23600	23200	22900	22500	22100	21800	21500	21200	20500	20000	19450	18950	18450	18000	17550	17150	16750
24	32	28000	27500	27000	26500	26100	25700	25300	24900	24500	24100	23700	23400	23100	22400	21800	21200	20600	20100	19600	19150	18700	18200
26	32	32900	32300	31700	31200	30700	30200	29700	29200	28700	28300	27900	27500	27100	26300	25600	24900	24200	23600	23000	22400	21900	21400
28	32	38100	37400	36800	36200	35600	35000	34400	33900	33400	32800	32300	31800	31400	30500	29700	28800	28100	27400	26700	26000	25400	24800
30	32	43600	42900	42200	41400	40700	40000	39400	38800	38200	37600	37000	36500	35900	34900	34000	33100	32200	31400	30600	29800	29200	28500
20	**34**	**20700**	**20300**	**19950**	**19600**	**19300**	**18950**	**18700**	**18400**	**18100**	**17800**	**17500**	**17250**	**17000**	**16500**	**16050**	**15650**	**15250**	**14800**	**14400**	**14100**	**13750**	**13400**
21	34	22800	22400	22000	21600	21200	20900	20600	20300	19900	19600	19300	19000	18750	18200	17750	17250	16800	16350	15900	15500	15150	14800
22	34	25000	24600	24200	23700	23300	23000	22600	22200	21900	21500	21200	20900	20600	20000	19450	18900	18450	17950	17450	17000	16600	16250
24	34	29800	29200	28700	28200	27800	27400	26800	26400	26000	25700	25300	24800	24500	23800	23200	22500	21900	21300	20800	20300	19800	19400
26	34	34900	34300	33700	33200	32600	32100	31600	31100	30600	30100	29700	29200	28700	27900	27200	26400	25800	25100	24400	23800	23300	22700
27	34	37700	37000	36300	35800	35200	34600	34000	33400	32900	32400	32000	31500	31000	30200	29300	28500	27800	27000	26300	25700	25100	24600
28	34	40400	39700	39100	38400	37800	37200	36500	36000	35400	34800	34400	33800	33300	32400	31500	30500	29700	29000	28300	27600	26900	26300
29	34	43400	42600	41900	41200	40500	39800	39200	38500	37900	37400	36800	36200	35700	34700	33800	32800	32000	31200	30400	29600	28900	28200
30	34	46500	45600	44900	44200	43400	42700	42000	41300	40700	40000	39400	38800	38300	37200	36100	35100	34200	33300	32500	31700	31000	30200

Comparison of Results.—In Fig. 13 are plotted tractive force or pull-speed curves of the same locomotive, based on the different methods discussed in this chapter. The curves are for an Atlantic type superheater locomotive of the E6s class, with cylinders 22 in. diam. and 26 in. stroke, drivers 80 in. diam. and total heating surface (based on water side of tubes) including superheater 3348 sq. ft., these being the dimensions for the older type of this class engine and the same as the one tried out on the locomotive testing plant. The new type E6s gives better results throughout.

The curve "*T.P.*" gives test plant observations for the pull behind the locomotive, plotted from the average of best results.

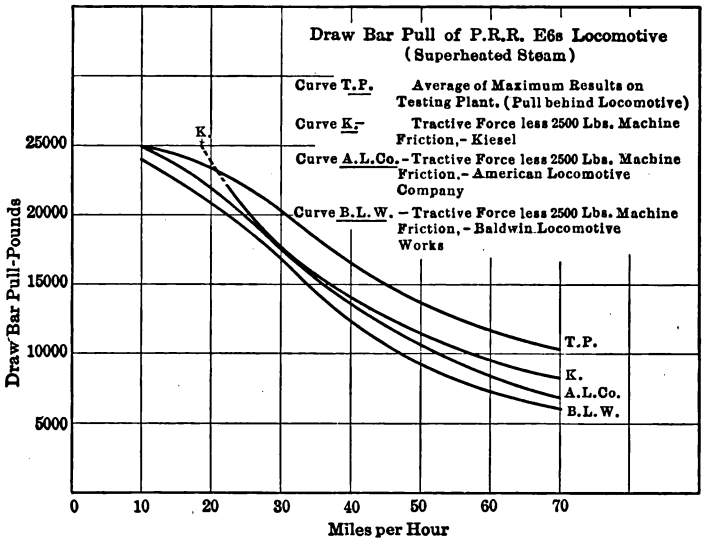


FIG. 13.—Comparison of tractive force formulas, applied to an Atlantic type locomotive.

The average machine friction was found to be 2500 lb. and this amount has been subtracted from each of the values calculated by the three methods noted for the cylinder tractive force at different speeds, thus reducing all results to a common basis for comparison.

For this locomotive, Kiesel's values by Eq. 12, reduce to the following form:

$$T.F. = \frac{2 \times 195 \times 157.3}{1 + \frac{110 \times 0.39}{3} \times \frac{157.3}{3348 \times 10} V} = \frac{61386}{1 + 0.0672V}$$

Where V = velocity, *m.p.h.*

This formula is figured on the basis of evaporation of 10 lb. water per sq. ft. total heating surface. To make a more accurate comparison with test plant results, a somewhat higher evaporation may consistently be used. The curves for the American Locomotive Company and the Baldwin Locomotive Works have been plotted using superheated steam factors as previously explained.

All formulas are based on service conditions. When locomotives are tested in a modern test plant under favorable conditions, they should be capable of developing from 10 to 15 per cent. more power than indicated by the formulas.

The Kiesel results apply to conditions where the locomotive is kept in repairs and worked under favorable conditions on the road. The relative positions of curves similar to those shown in Fig. 13, vary considerably with different classes of engines. The case here discussed is taken merely to call attention to the differences for this locomotive.

Train loading calculated by the Kiesel formula has been found in a large range of cases to agree with results of road trials.

Indicated Horse Power.—If V is the speed in miles per hour and T the theoretical tractive force,

$$T \times V \times 5280 = \text{i.h.p.} \times 33{,}000 \times 60$$

or

$$\text{i.h.p.} = \frac{T \times V}{375} \tag{13}$$

Since, from Eq. 9, $T = \frac{Pd^2s}{D}$, we may also write

$$\text{i.h.p.} = \frac{Pd^2sV}{375D} \tag{14}$$

which is the cylinder indicated horse power up to what the author prefers to call "steam stroke speeds," that is, up to speeds at which the boiler can no longer supply steam at full stroke.[1] The available horse power at the rim of the drivers is found by multiplying Eq. 13 or 14 by the factor .85. It is recommended that the horse power factors, as given in Fig. 12, be used in problems.

[1] See Eq. 97, page 222.

The American Locomotive Company has used the following for determining the maximum horse power:

$$\text{hp.} = 0.0212 \times P \times A \text{ for saturated steam} \quad (15)$$

$$\text{hp.} = 0.0229 \times P \times A \text{ for superheated steam} \quad (16)$$

A = area of one cylinder, sq. in.

P = boiler pressure, lb. per sq. in.

Equation 16 gives nearly 10 per cent. higher results than Eq. 15. It is of importance to note that one marked advantage of superheated steam over saturated steam lies in the sustained horse power for speeds corresponding to piston speeds over 1000 ft. per minute as shown in Fig. 12.

Problem 4.—(*a*) For the K2 locomotive, draw on a cross-section sheet, curves for the rated tractive force up to 60 m.p.h. locating point for each 10-mile interval, using (1) Baldwin, (2) American, (3) Kiesel formulas. For dimensions, see Table II.

(*b*) For the same locomotive as in (*a*) draw on the same sheet, curve for the i.h.p. using points for the American curve for T (tractive force) in Eq. 13 and locate points for 10-mile intervals.

QUESTIONS ON CHAPTERS I, II AND III

1. Does the horse power per minute equal numerically the horse power per hour?

2. What controls the amount of steam which enters the locomotive cylinders?

3. Why is it not feasible to apply a fly-wheel to a locomotive? Analyze the conditions in this particular, comparing them with a stationary engine.

4. What is the advantage of an Atlantic type over an American type locomotive? What class of service is best adapted to Consolidation locomotives?

5. Without referring to Fig. 5, name 30 parts of the locomotive shown in Fig. 2.

6. Steam pushes back on the piston with the same force as it pushes forward on the front cylinder head; how is it possible for the locomotive to move under this condition?

7. Show by sketches why the vertical component of piston pressure is on the upper guide when the locomotive is going ahead and on the lower when moving backward.

8. Distinguish between tractive force and rotative effort.

9. Why does the tractive force decrease with increase in speed?

10. By Table 3 and Fig. 11, find (*a*) the rated tractive force and (*b*) the tractive force at 40 m.h.p. for each of the locomotives in Table II.

CHAPTER IV

ACCELERATION OF TRAINS

REVIEW OF ELEMENTARY PRINCIPLES; UNIFORM AND VARIABLE ACCELERATION; A GENERAL CASE SOLVED; CURVES FROM TESTS; HOW TO CORRECT FOR ACCELERATION

The basis for solving problems involving a change of speed, may be stated as follows: A constant moving force F acts on a mass M producing a rate of change of velocity a. The acceleration, a, may be expressed in terms of other units, as space and time. The fundamental equation may be written:

$$F = Ma = \frac{W}{g}\,a = \frac{Wa}{32.16} \text{ or } a = \frac{gF}{W} = \frac{32.16F}{W} \qquad (17)$$

in which F is expressed in pounds or its equivalent (as pounds per ton) W, the weight moved, must also be in the corresponding units and, a, acceleration (or deceleration) in feet per second per second when 32.16 appears in the equation without a factor to change it to any other acceleration unit.[1]

If the acceleration is uniform, it is measured by the amount by which the velocity, v, is increased in a unit of time t, or

$$a = \frac{v}{t} \qquad (17a)$$

If the acceleration is variable, it is measured by the amount by which the velocity would be increased in a unit of time if its rate of increase continued the same as at the instant considered. It may be expressed by the differential equations,

$$a = \frac{dv}{dt} = \frac{d^2S}{dt^2} \qquad (17b)$$

where v and t are the same as used above and S is the space passed over in the time t.

When the acceleration is uniform, the relations between the

[1] The reader is cautioned not to pass over hastily or carlessly these fundamental conceptions. They underlie the entire theory of train operation.

time and velocity may be shown by a triangle, as Fig. 14. Let AB represent the time, say 3 sec. Divide it into three equal spaces, and each space will represent a second. Draw horizontal lines through the points of division and limit them by the inclined line AC. The horizontal lines will represent the corresponding velocities. Thus $v_2 = de$ is the velocity at the end of the time t_2. The *triangle Abc* represents the *space* passed over during the first second, and ABC the space passed over during 3 sec. The lines ge and fC represent the acceleration for successive seconds, which in this case equal each other and equal bc which is the velocity at the end of the first second. Hence, when the acceleration is uniform, the velocity at the end of the first second equals the acceleration.

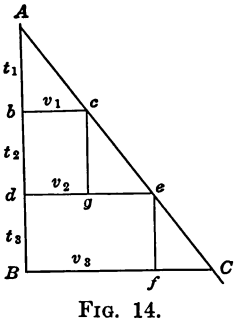


FIG. 14.

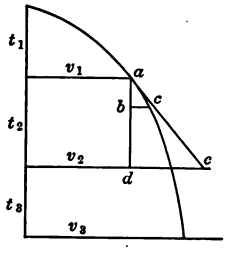


FIG. 15.

If acceleration constantly varies, the case may be represented by Fig. 15. To find the acceleration at the end of the first second, draw a tangent ae to the curve at the point a, and drop the perpendicular ad, then will de be the acceleration. But $\frac{de}{ad} = \frac{bc}{ab} = \frac{dv}{dt} = a$ = the velocity-increment or the relation of the rate of change of velocity; ab represents an increment of the time and bc an increment of the velocity.

By referring to Fig. 14, we see that if the initial velocity is zero, $S = \frac{1}{2}vt$, but since $v = at$

$$S = \tfrac{1}{2}\, at^2 \text{ or } a = \frac{v^2}{2S} \qquad (18)$$

The force necessary to produce this acceleration is

$$F = \frac{w}{32.16} \times \frac{v^2}{2S} = \frac{wv^2}{64.32S}$$

If the velocity is in miles per hour $v = \frac{5280V}{60 \times 60} = 1.466V$ or

$$F = \frac{w}{64.32S} \times (1.466V)^2$$

If W = tons of 2000 lb., $w = 2000W$

$$F = \frac{2000W \times 2.15V^2}{64.32S} = \frac{66.8WV^2}{S} \tag{19}$$

That is, F is the force in pounds required to accelerate a train weighing W tons from rest to a velocity V m.p.h. in a distance S ft. But in addition to this, a certain force is required to set the wheels rotating to attain this velocity and this has been found to vary from 4 to 7 per cent. of Eq. 19. Taking the average as 5 per cent., we have finally

$$F = \frac{66.8WV^2}{S} \times 1.05 = \frac{70V^2W}{S} \text{ (nearly)} \tag{19a}$$

F expressed in pounds per ton, becomes

$$F_t = \frac{70V^2}{S} \tag{20}$$

Introducing two velocities,

$$F_t = \frac{70(V_2^2 - V_1^2)}{S} \tag{20a}$$

Again introducing W, we may write

$$F = 70 \frac{W}{S} (V_2^2 - V_1^2) \tag{21}$$

where V_2 is the higher and V_1 the lower velocity. This equation may also be written in the following forms:

$$S = 70 \frac{W}{F} (V_2^2 - V_1^2) \tag{22}$$

$$W = \frac{FS}{70(V_2^2 - V_1^2)} \tag{23}$$

$$V_2 = \sqrt{\frac{FS}{70W} + V_1^2} \tag{24}$$

Going back to Eq. 17,

$$a = \frac{32.16F}{W}$$

if a is expressed in miles per hour per second and W is short tons $= \frac{W}{2000}$,

$$a = \frac{F}{W} \times \frac{32.16 \times 3600}{1.466 \times 2000} \text{ or}$$

$$\frac{F}{W} = 91.1 \times \text{acceleration in m.p.h. per sec.},$$

which means that 91.1 lb. accelerating force will uniformly accelerate 1 ton at the rate of 1 m.p.h. in each second.

If effect of inertia of revolving wheels be added, $91.1 \times 1.05 = 95.65$ lb. total force for the accelerating or decelerating force. Therefore

$$F = 95.65aW = 95.65 \frac{V_2 - V_1}{t} W \tag{25}$$

or in pounds per ton,

$$F_t = 95.65 \frac{V_2 - V_1}{t} \tag{25a}$$

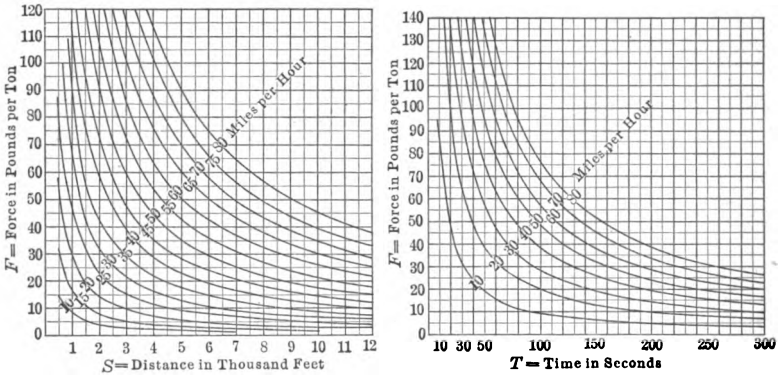


FIG. 16.—Graphical representation of Eq. 20a and 25a.

Equations 20a and 25a are plotted in Fig. 16 and apply to acceleration and deceleration providing the rate of change of velocity is uniform between any two velocities considered.

Approximate Formulas.—It may be found convenient, where approximate calculations only are desired, to apply the following formulas for uniform acceleration (or deceleration), the calculations to be made at 5-sec. intervals during the change of speed. Let:

$T.F.$ = tractive force in pounds per ton.
Acceleration = rate of change of velocity in miles per hour per second.
V_1 = initial speed, miles per hour.
V_2 = final speed, miles per hour.
S = distance run in feet.
t = time in seconds to run S ft.
Friction = sum of all resistances, other than acceleration resistances, in pounds per ton.

As formerly deduced, approximately 96 lb. of tractive force will uniformly accelerate 1 ton at the rate of 1 m.p.h. per sec. on a level track, including inertia of revolving wheels.

$$T.F. = \text{Acceleration} \times 96 + \text{friction}$$

Suppose a train reaches a speed of 30 m.p.h. in 3 minutes, the acceleration is $\frac{V}{t} = \frac{30}{3 \times 60} = \frac{1}{6}$ m.p.h. per sec. and the equivalent $T.F. = \frac{96}{6} = 16$ lb. per ton. This is the force to accelerate the train and includes the effect of inertia of the revolving wheels.

$$S = t^2 \times (T.F. - \text{friction}) \times 0.00763$$

or $$S = t \times V_2 \times 0.733$$

or $$S = \frac{V_2^2 \times 70}{T.F. - \text{friction}} \tag{26}$$

$$V_2 = t \times (T.F. - \text{friction})$$

or $$V_2 = \frac{S}{t \times 0.733}$$

$$t = \frac{S}{V_2 \times 0.733} \tag{27}$$

$$T.F. = \frac{S}{t^2 \times 0.00763} + \text{friction} \tag{28}$$

$$T.F. = \frac{V_2^2 \times 70}{S} + \text{friction} \tag{29}$$

Problem 5.—If a train on a level track is accelerated from rest to 50 m.p.h. in 10 minutes, find the (1) rate of acceleration in miles per hour per second, and (2) the distance run without friction, and (3) the equivalent *T.F.* for the determined rate of acceleration.

A Special Case.—As stated before, the above formulas apply only to problems involving uniform rate of change of velocity. The drawbar pull continues to decrease as the speed increases; while, on the other hand, the train resistance increases as the speed increases and at some "balancing speed" absorbs all of the energy developed by the motive power. For these reasons, to assume uniform acceleration for any considerable distance is but a crude approximation.

By referring to Fig. 18, it will be observed that a train is accelerated uniformly for but a short distance, the speed-time curves 5 and 7 being straight for a limited distance only. Such curves may often be represented by an equation of the general form:

$$v^2 = as - bs^2 \tag{30}$$

in which v is the velocity in feet per second, s the distance run in feet to reach a velocity v, a and b are constants; but,

$$v = \frac{ds}{dt} \text{ or } \frac{ds^2}{dt^2} = as - bs^2$$

Differentiating, $$2ds\,\frac{d^2s}{dt^2} = ads - 2bsds$$

Divide through by $2ds$, we obtain

$$\frac{d^2s}{dt^2} = \frac{1}{2}a - bs \text{ but } \frac{d^2s}{dt^2} \text{ is the acceleration (or deceleration)}$$

Hence $$F = \frac{W}{g} \times \frac{d^2s}{dt^2} = \frac{W}{32.16}\left(\frac{1}{2}a - bs\right) \tag{30a}$$

Extending further the mathematical applications to an actual case: Let the axes $O'X$ and $O'Y$ (Fig. 17) represent the asymptotes to the "speed-time" curve, OP, the train accelerating from O to P.

Theory Applied.—Assume curve is an equilateral hyperbola, equation of which is $y = \frac{a'}{x}$, a' being an unknown constant.

$x = c + t$; $y = -(b - v)$, so that equation of the velocity curve referred to the axis OX and OT becomes

$$v = \frac{a' + bc + bt}{c + t}$$

When $t = o$, v must be o, and this requires $a' + bc = o$.

Therefore $$v = \frac{bt}{c+t}$$

The acceleration, p, in terms of time:

$$p = \frac{dv}{dt} = \frac{bc}{(c+t)^2}.$$

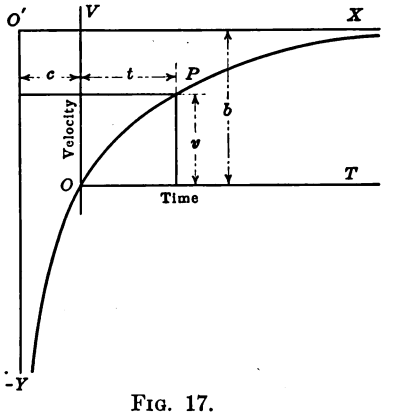


FIG. 17.

Applying this in terms of the differential equation:

$$\frac{ds}{dt} = v = \frac{bt}{c+t} = \frac{-bc}{c+t} + \frac{b(c+t)}{c+t} = b - \frac{bc}{c+t}.$$

$$\int_o^s ds = \int_o^t \left\{ bdt - \frac{bc}{c+t} dt \right\}$$

$$s = bt - bc \text{ hyp log } (c+t) + bc \text{ hyp log } c$$

$$= bt - bc\{\text{hyp log } (c+t) - \text{hyp log } c\}$$

$$= bt - bc \text{ hyp log } \frac{c+t}{c} \qquad (31)$$

The equations for determining b and c (from method of "least squares")

$$\Sigma\{vtc + vt^2 - t^2b\} = o$$
$$\Sigma\{v^2c + v^2t - vtb\} = o$$

The above method has been applied to the following conditions, taken from observed data:

for $t =$	5	$v = 28.0$
	10	44.0
	15	51.5
	20	56.0
	25	60.5

From this data, $b = 82.043$; $c = 9.011$

$$v = \frac{82.043t}{9.011 + t} \qquad p = \frac{739.289}{(9.011 + t)^2}$$

$$s = 82.042t - 1702.3 \log \frac{9.011 + t}{9.011}$$

The calculated values of v agree well with observed values, as shown in the following:

t	Calculated v	Observed v	Error
5	29.3	28.0	+1.3
10	43.2	44.0	−0.8
15	51.3	51.5	−0.2
20	56.6	56.0	−0.6
25	60.3	60.5	−0.2

Curves from Tests.—Fig. 18 presents a study of the practical application of the theory of acceleration. It is a dynamometer car record of the pull on a Pacific type locomotive in starting 10 steel cars from rest and accelerating the train on a nearly level roadbed. Before the original dynamometer car record, shown in curve 1, can be satisfactorily studied independently for acceleration resistance, it is necessary to subtract from the actual drawbar pull all resistances other than the one for a change in speed. The plot shows the following curves:

1. Actual drawbar pull.
2. Corrected drawbar pull.
3. Capacity drawbar pull.
4. Mean (actual) drawbar pull.
5. Speed, miles per hour.
6. Distance curve.
7. Speed, feet per second.

The corrected drawbar pull curve is plotted from the actual drawbar pull curve after correcting for the effect of grade and acceleration and which may be explained by assuming the following conditions, taken directly from Fig. 18. Weight of locomotive and tender 215 tons on a grade of −0.62 per cent.

Time from start, seconds	Speed, feet per second	Actual drawbar pull, pounds
50	30.2	.....
55	32.7	16,600
60	35.5	.....

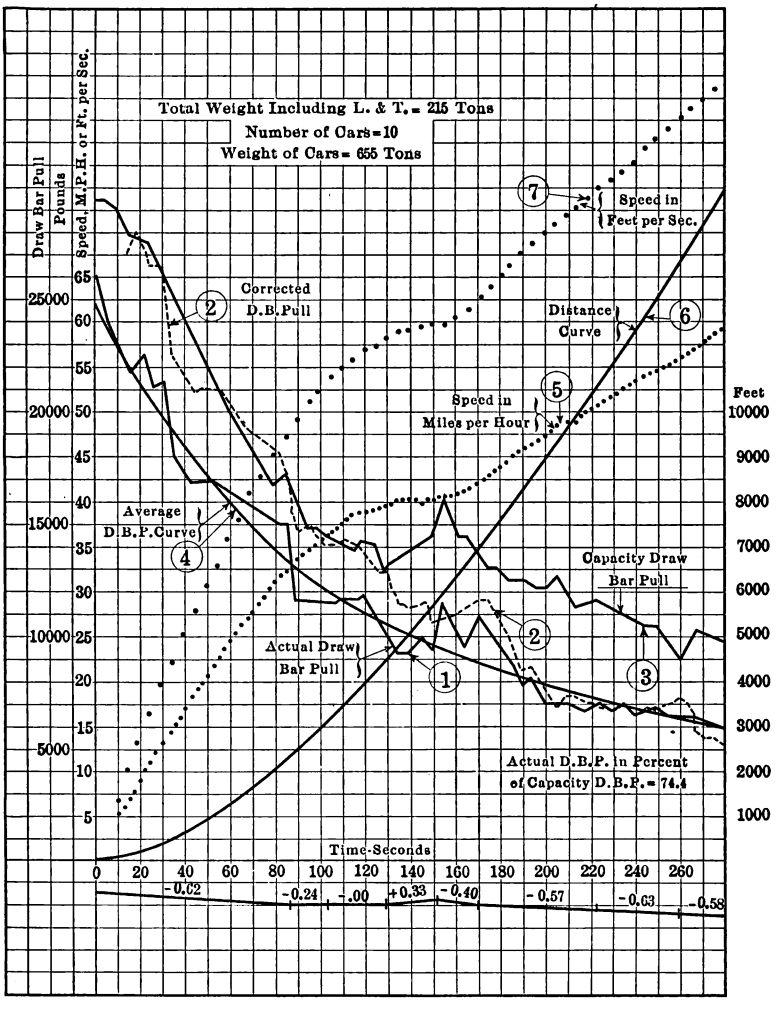

Total Weight Including L. & T. = 215 Tons
Number of Cars = 10
Weight of Cars = 655 Tons
Draw Bar Pull Pounds
Speed, M.P.H. or Ft. per Sec.
Corrected D.B.Pull
Average D.B.P.Curve
Actual Draw Bar Pull
Capacity Draw Bar Pull
Speed in Feet per Sec.
Distance Curve
Speed in Miles per Hour
Feet
Actual D.B.P. in Percent of Capacity D.B.P. = 74.4
Time-Seconds
0 20 40 60 80 100 120 140 160 180 200 220 240 260
-0.62 -0.24 -.00 +0.33 -0.40 -0.57 -0.63 -0.58

FIG. 18.—A study of acceleration tests.

The average rate of acceleration = 0.53 ft. per sec. per sec.

The force required to accelerate the locomotive and tender at the above rate is:

$$F = \frac{Wa}{32.16}; F_t = 62.2a, \text{ where}$$

F_t = accelerating force, pounds per ton.

or

$$F = 62.2 \times 215 \times 0.53 = 6860 \text{ lb.}$$

Since the actual drawbar pull does not measure either the acceleration or the grade effect due to locomotive and tender, curve 1, Fig. 18, must be corrected as follows:

(*a*) The accelerating force found above must be added to the actual drawbar pull, since this pull is part of the total force exerted by the steam on the piston.

(*b*) The effect of down-grade is to require less accelerating force than would be required on a level track. As derived on page 48, each per cent. of up-grade offers a resistance of 20 lbs. per ton. Therefore, the effect of the grade in question, being negative, will be subtracted.

$$-0.62 \times 20 \times 215 = -2666 \text{ lbs.}$$

The corrected drawbar pull will therefore be,

$$16{,}600 + 6860 - 2666 = 20{,}794$$

This is the tractive effort which the locomotive would have exerted at constant speed on level track had it been worked as when the above drawbar pull (16,600) was obtained.

The capacity drawbar pull is obtained as follows: A K2 locomotive, similar to the one used in obtaining the results in Fig. 18, can exert at a speed of 32.7 ft. per sec., a drawbar pull of 25,300 lb. If this locomotive was being worked to its limit and accelerating at the same rate and on the same grade as the case just considered, then would the actual drawbar pull be,

$$25{,}300 - 6860 + 2266 = 20{,}700$$

instead of 16,600 as in the example worked.

Referring to curves of this character, Mr. R. D. Kavanaugh, Test Department, Pennsylvania Railroad, states that in making up drawbar pull curves for locomotives in road tests, corrected drawbar pull points are plotted only after a large number of tests have been made and with assurance that the points were obtained under maximum working conditions of the locomotive. In this way, a curve may be drawn which usually passes through the higher points thus obtained. If such data is not available,

approximate results may be obtained by applying Eq. 11 to the speed conditions.

Problem 6.—Check the "corrected" and the "capacity" curves, Fig. 18, for the curves from 60 to 70 sec., following the method shown above and using a K2 locomotive, as given in Table II, page 12. To determine what may be expected of this locomotive at the speed corresponding to the average time of 65 seconds, apply Eq. 11, using the proper speed factor.

Problem 7.—Calculate the drawbar pull by applying Eq. 30*a* to the conditions represented in Fig. 18, for speeds of 5, 15, 20, and 30 ft. per sec. (Hint: To obtain the constants *a* and *b*, select say four points along the nearly straight part of curve 7 and substitute in Eq. 30, the values of v corresponding to s, giving four independent equations, which may be solved for *a* and *b*.)

The correction required for acceleration may be obtained directly from Eq. 21 in space units and from Eq. 25 in time units; or, values may be read directly from Fig. 16, observing that W is there expressed in pounds per ton. The author prefers the following derivation: Let

F = correction in pounds.
F_t = correction in pounds per ton.
W = weight of car in tons.
w = weight of wheels under car.
v_1 = initial velocity, feet per second.
v_2 = final velocity, feet per second.
S = distance run during acceleration (or retardation) between velocity change from v_1 to v_2.

Work $= FS = \frac{1}{2} Mv^2$

$$F = \frac{W + 0.6w}{2 \times 32.16} \times 2000 \times \frac{(v_2 + v_1)(v_2 - v_1)}{S}$$

$$F = W\left(1 + \frac{0.6w}{W}\right) \times 31.05 \times \frac{(v_2 + v_1)(v_2 - v_1)}{S} \qquad (32)$$

If t = time in seconds to change from v_2 to v_1, substitute $\frac{v_2 + v_1}{2} = \frac{S}{t}$ and multiply by 1.466 to change to V, in miles per hour

$$F = \frac{91.1}{t}(V_2 - V_1)\left(1 + \frac{0.6w}{W}\right)W \qquad (33)$$

For T tons in total train of N cars, $\frac{T}{W} = N$. Multiply Eq. 33 by T and substitute $\frac{T}{W} = N$

Therefore:

$$F_t = \frac{F}{W} = \frac{91.1}{t}(V_2 - V_1)(T + 0.6wN)$$

Since $0.6w = 1.68$ tons per car for 33 in. wheels and $5\frac{1}{2} \times 10$ in. axles,

$$F_t = \frac{91.1}{t}(V_2 - V_1)(T + 1.68N) \qquad (34)$$

Where F is positive, its value should be subtracted from the drawbar pull and when negative it should be added to the drawbar pull.

If greater accuracy is required, the inertia effect of the revolving wheels should be considered and the above should be multiplied by a factor, an average value for which is 1.05, giving a constant 95.65 as in Eq. 25, instead of 91.1, and we may write, finally:

$$F = \frac{95.65}{t}(V_2 - V_1)(T + 1.68N) \qquad (35)$$

Problem 8.—Find the correction for acceleration in Fig. 18 for the ten cars in that train in accelerating from 10 to 30 m.p.h. Plot the acceleration curve up to 30 m.p.h. on a sheet of cross-section paper, assuming the profile Fig. 18 is perfectly level.

CHAPTER V

TRAIN RESISTANCE

Speed, Grade, Acceleration, Curve, Flange, Internal and Other Resistances

Train resistance may be defined as the sum of all those forces which act unfavorably to retard the movement of a train. It has two main divisions (*A*) Frictional Resistances and (*B*) Grade and Acceleration Resistances.

Brake shoe friction and the friction of wheels on the track would not be classed under either of the above, since both are favorable (or useful) resistances. Without track friction the locomotive could not move the train nor could the cars be braked effectively; without brake resistance train operation would not be safe.

Speed resistance is a general term for the frictional resistances and under (*A*) is also included machine friction of the locomotive driving mechanism. Machine friction includes frictional resistance offered by: (1) Piston-rod and stuffing box, (2) cross-head guide and pin, (3) main pin bearings, (4) driving axle journals, (5) side rod bearings, (6) link motion, (7) valves and valve rods.

The earliest attempt to determine train resistance was made by Stephenson and Wood in 1818. In 1855, D. K. Clark proposed for total speed resistance the values expressed by $R = 7.2 + 0.0053\ V^2$ where R is the resistance in pounds per ton of 2000 lb. and V the velocity in miles per hour. The problem of expressing the various frictional resistances by means of simple equations has attracted the attention of many engineers and as traffic has developed with its attending increase in speeds and loading, the earlier values have been found much too high for speeds above 40 m.p.h. Among the later formulas are

$$R = 2 + \frac{V}{4} \text{ (Engineering News)}$$

$$R = 3 + \frac{V}{6} \text{ (Baldwin Locomotive Works)}$$

in which R is the resistance in pounds per ton and V the speed in miles per hour. These give values some higher than accepted to-day but are approximately correct for passenger cars. They do not take account of variations due to loading and do not apply to freight cars nor to locomotives.

Before presenting present day results, we may consider some of the conditions which have an influence on the resistance as a whole.

A. Condition of Cars.

1. Weight of car. Resistance per ton is less with heavy cars than with light cars.
2. Arrangement and number of wheels in trucks.
3. Squareness of trucks or axles.
4. Diameter of wheels and journals.
5. Wheel base of trucks and cars.
6. Condition and character of centerplates and side bearings.
7. Kind of lubrication, including journal friction.
8. Materials of journals and journal bearings.
9. Speed of train. At speeds above 5 or 6 m.p.h., resistance will increase with the speed. In starting, the friction is three or four times as much as at 5 m.p.h.

B. Track Conditions.

1. Surfacing of track in horizontal plane.
2. Alignment of track.
3. Rigidity of track, especially at joints.
4. Gage of track or endplay between rail and wheels.
5. Curvature, degree and length of curve and super-elevation used; where curve is compensated, amount of compensation.
6. Grade, per cent. and length.
7. Where calculations are made using track profiles, accuracy of profiles.

C. Weather Conditions.

1. Temperature: the locomotive drawbar pull decreases and car resistance increases with low temperature.
2. Wind which will reduce the locomotive drawbar pull and will increase car resistance.

Mr. F. J. Cole, Chief Consulting Engineer, American Locomotive Co., has made the following divisions of the general subject.[1]

[1] Railroad Age Gazette, August 27, 1909.

- Train resistance
  - Acceleration resistance depends on tonnage only. — 20 lb. per ton for 1 per cent. grade.
  - Grade resistance depends on tonnage only. — 20 lb. per ton for 1 per cent. grade.
  - Friction — Varies from 2½ to 12 lb. per ton.
    - Journal, depends on design and lubrication.
    - Rolling, depends on solidity of roadbed.
    - Flange.
    - Curve
      - Sliding of one tread in each pair, on the rail.
      - Flange friction
    - Side bearings and center plates cause flange friction.
    - Wind
      - Wind against body of car.
      - Flange friction.

Flange friction consists of that due to curves, side bearings and center plates, winds, etc.

Acceleration Resistance.—In addition to the forces to overcome all other resistances, a certain amount of pull is required to accelerate a train from a lower to a higher velocity. Chap. IV (page 43) discusses the principles on which this resistance is based and how it should be applied in problems. From Eq. 20*a*

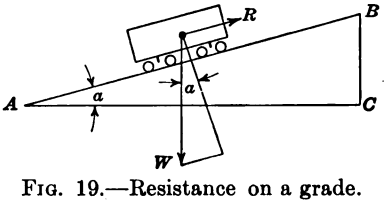


FIG. 19.—Resistance on a grade.

$$F_t = \frac{70\ (V_2^2 - V_1^2)}{S} \tag{36}$$

Where F_t becomes the force to overcome the resistance opposed to accelerating, in pounds per ton, V_2 the higher and V_1 lower velocity in miles per hour and S the distance run in feet during the change in velocity. As noted, this equation is to be used only for *uniformly increasing* or *decreasing rates* of *velocity*. This equation is plotted in Fig. 16.

Grade resistance is the resistance required to overcome the force of gravity in lifting the total weight of the train through the height of the grade. It is a positive quantity for an up-grade and a negative quantity for a down-grade. Since the work of lifting the weight W a distance represented by BC, Fig. 19, while

ascending a grade AB is equal to the work required to overcome the resistance R (due to grade only) through a distance AB,

$$W \times BC = R \times AB$$

or

$$R = W \times \frac{\text{rise}}{\text{length of incline}}$$

$R = W \times \sin a$ where a is the angle of the grade.

Since the angle a is small, we may write

$$R = W \tan a$$

But $\tan a = \frac{CB}{AC} = \frac{G}{100}$. Where G is the rise per 100 ft., or the per cent. of grade.

Hence, $R = W \frac{G}{100}$. If R is the resistance in pounds per ton of weight of train we have,

$$R = \frac{2000G}{100}$$

or

$$R = 20G \tag{37}$$

Hence the resistance due to grade in pounds per ton is found by multiplying the per cent. of grade by 20.

This is independent of other resistances. Wherever there is a grade, it is to be added to other resistances when the train is ascending and subtracted from other resistances when descending a grade. To find the total resistance due to grade in pounds per ton of 2000 lb., multiply the rise in feet per mile by 0.3788.

Virtual or Velocity Grade.—Suppose a train approaches a 1.0 per cent. up-grade 10,000 ft. long at 40 m.p.h. and the speed is reduced in that distance to 10 m.p.h.

Using the values above in Eq. 36 we have the accelerating force assisting the engine,

$$70 \times \frac{1600 - 100}{10000} = 10.50 \text{ lb. per ton}$$

The resistance of the 1.0 per cent. grade is 20 lb. per ton; then, $1 - \frac{10.5}{20} = 0.475$ = virtual grade in per cent. *The virtual or velocity grade is (in the above case) the actual grade less a grade which is equivalent in effect to the acceleration resistance corresponding to the change in velocity.*

The above method applies equally to the case of a locomotive increasing the speed of its train on a grade, in which case it overcomes the train resistance, the grade resistance and also the resistance of a grade equivalent in effect to the acceleration resistance. The virtual grade would be the actual grade *plus* the grade of acceleration. *In general, the virtual grade is the actual grade corrected for the effect (due to change of velocity) of the momentum of the moving train.*

Machine Friction.—This refers to the internal resistances of the locomotive. It has been common practice to assume its value as 22½ lb. per ton of weight on drivers, but this appears to be somewhat low. An Atlantic type locomotive under plant test gave 31.8 lb. per ton at 47 m.p.h. which increased to 41 lb. at 75 m.p.h. A Pacific type locomotive gave at low speeds 23.3 lb. per ton weight on drivers which decreased to a minimum of 12.05 lb. per ton at 37.8 m.p.h. and reached 39.8 lb. per ton at 85.1 m.p.h.

Some years ago, Prof. Goss proposed the following for internal resistance, $3.8 \frac{d^2 l}{D}$. In the Atlantic type locomotive just referred to, this would give $3.8 \times \frac{22^2 \times 26}{80} = 598$ lb., which does not check favorably with the tests referred to. Kiesel's formula, Eq. 44, gives results based upon weight on drivers, number of drivers and the speed. Test plant results for frictional resistance do not check with any of the commonly used formulas, the same engine varying widely with a slight variation in conditions under which it may be run.

Curve Resistance.—When a four-wheel truck with a long wheel base running on a straight track hits a curve, the forward outer flange meets a resistance, tending to skew the truck across the track and this action on the truck is resisted by a normal pressure against the inner flange on the inner rail. The truck is therefore guided around the curve by the action of the force on the outer rail.

One of the most satisfactory analyses, based on theory, treating of the action of a wheel on a curve, is by Dean Raymond,[1] who deduces the following expression, which shows to what extent wheel base and curvature modify average results.

[1] This analysis may be extended to include determination of maximum safe speed, as given in Railroad Gazette, Vol. XLIV, No. 11.

$$R_c = 0.4 + (0.21 + 0.035a)D. \qquad (38)$$

R_c = curve resistance, pound per ton of total load.
a = distance in feet between axle centers of truck.
D = curvature in degrees.

Wheel base $= a$	Resistance, pounds per ton $= R_c$
5	$0.4 + 0.385\ D$
6	$0.4 + 0.420\ D$
7	$0.4 + 0.465\ D$
8	$0.4 + 0.490\ D$
9	$0.4 + 0.525\ D$
12	$0.4 + 0.630\ D$
13	$0.4 + 0.665\ D$
15	$0.4 + 0.735\ D$
16	$0.4 + 0.770\ D$
20	$0.4 + 0.910\ D$

In rounding curves, it is necessary to determine the proper superelevation of the outer rail, thus reducing flange resistance.[1]

Let G = gage of track in feet (assumed in tables as = 4 ft. 9 in.).
E = superelevation of outer rail in feet.
E_1 = superelevation of outer rail in inches.
V = velocity of train in feet per second.
V_1 = velocity of train in miles per hour.
F = centrifugal force in pounds.
W = weight of locomotive in pounds.
R = radius of curve in feet.
H = height of center of gravity of locomotive in feet.

Construct Fig. 20 so that df shall represent the gage G of the track, fe the superelevation E; bc the horizontal line from the middle point c of the line df will represent the centrifugal force F, and the perpendicular ab the weight W of the locomotive, such that the resultant shall be perpendicular to the track and shall pass through the center point c.

In the similar right triangles abc and def, we have

$$\text{or} \qquad ab : bc :: de\ (= df, \text{ approx.}) : ef$$
$$W : F :: G : E$$

[1] See "The Elements of Railroad Engineering" by Wm. G. Raymond.

or

$$E = F\frac{G}{W}$$

but

$$F = \frac{W}{32.16} \times \frac{V^2}{R} \; (= \text{Centrifugal force})$$

Substituting, we have

$$E = \frac{W}{32.16} \times \frac{V^2}{R} \times \frac{G}{W} = V^2 \frac{G}{32.16R}$$

or

$$V^2 = 32.16\, R \frac{E}{G}$$

or

$$V_1^2 = 1.2R\frac{E_1}{G} = R\frac{E_1}{4} \text{ when } G = 4.75$$

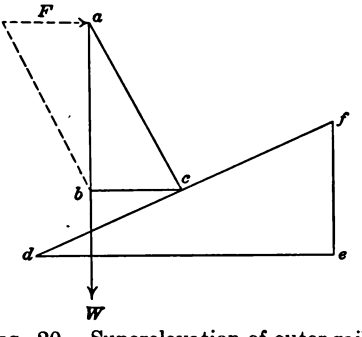


FIG. 20.—Superelevation of outer rail.

$$V_1 = \sqrt{R\frac{E_1}{4}} = \text{proper speed for a given elevation and curvature} \qquad (39)$$

$$E_1 = \frac{4V_1^2}{R} = \text{proper elevation for a given speed and curvature} \qquad (40)$$

Exercise Applying Eq. 40.—Find the proper elevation of the outer rail for a train moving at 20 m.p.h. about a curve of 1000 ft. radius. Same for 60 m.p.h. and 3000 ft. radius.

Pressure of the flange of the tire against the rail in curving produces flange wear. Fig. 21 shows the normal position of a chilled tread on an American Society of Civil Engineers' 100 lb. rail, the actual areas in contact being from ⅜ to ½ sq. in. The

total play between the gage of the wheels and the gage of the track is about 3/16 in., which if decreased, the wheel will lift from the rail. It may be four times this value for driving wheels. When the play is taken up, the flanges will wear, an extreme case being shown in the dotted lines, where the flange wear is vertical. When this wear exceeds the allowable limits fixed by standard rules of interchange, the tires must be turned down resulting in the waste of much good metal, the condition being illustrated in Fig. 21. This wear from rounding curves is the more serious on the leading truck wheels.[1]

The wear on the flange on a straight track is not excessive, but it may be a function of the speed. Cole has found from cal-

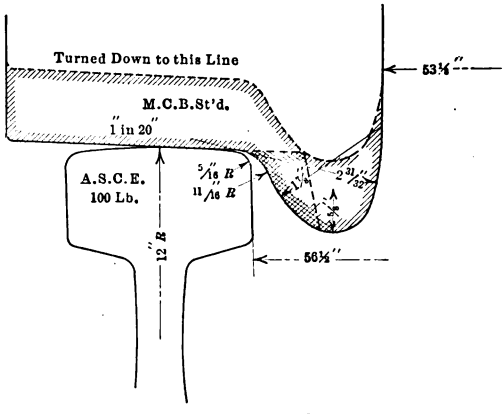


FIG. 21.—Tread before and after tire wear.

culations using the Carus-Wilson formula that at 10 m.p.h. on a car of 30 tons total weight, the flange action accounts for 29.3 per cent. of the total frictional resistance, the journal friction amounts to 63 per cent., rolling 4.2 per cent. and air resistance 3.5 per cent. The corresponding figure for 35 miles an hour is 48.5 per cent. for flange action, 29.5 for journal, 2 for rolling and 20 per cent. for air resistance.

The common allowance for curve resistance is from 0.8 to 1.2 lb. per ton per degree, *with an average of* 1.0. A study of various results shows values as low as 0.40 per ton per degree (Wellington for fast trains) to values as high as 2.0 estimated in starting on a curve.

[1] For experimental determination of amount of this thrust, see tests by Mr. George L. Fowler, Railroad Gazette, Nov. 15, 1907.

To express curve resistance in equivalent grade divide by 20. Thus, if the curvature is 1° and the resistance is 1 lb., the equivalent grade in per cent. would be 0.05.

One degree of curvature is equal to a radius of 5730 ft.

Curves on heavy grades often require that the grades be compensated for the curve; that is, reduce the grade at the curve by such an amount that the total train resistance due to the two will not exceed the grade resistance of the ruling grade on that division. Raymond suggests the following:

Degree of curve	1	2–4	5 and over
Grade compensated per degree	.04	0.30	.025

Atmospheric Resistance.—A fair average value from experiments by Goss for head air resistance is

$$R = 0.1\ V^2 \tag{41}$$

where R is the resistance in pounds per ton and V the velocity in miles per hour. Other values are

Engine and tender, $R = 0.11V^2$
First car of train, $R = 0.001V^2$
Last car of train, $R = 0.00026V^2$
Intermediate coach, $R = 0.0001V^2$

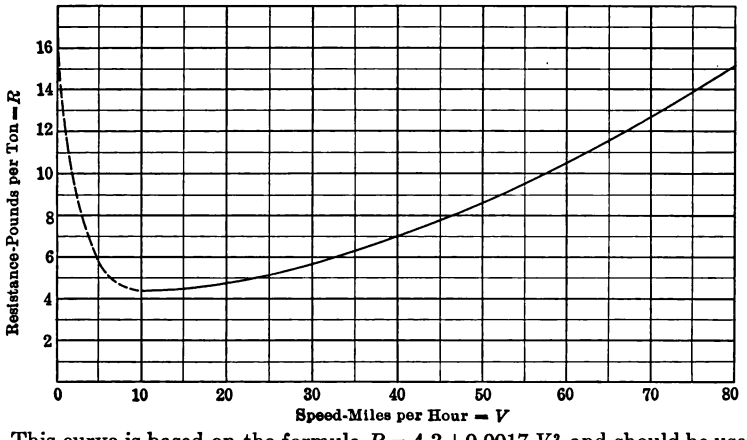
Effect of Cold Weather.—Experiments made in 1909 by Prof. E. C. Schmidt on the Illinois Central R. R. indicate that the frictional resistance of freight cars moving at 10 to 12 m.p.h. is 50 per cent. greater at a temperature of 0°F. than at 70°F. At 20 m.p.h. the above figure will be increased to 67 per cent.

In addition to the increase in train resistance, the resistance of the locomotive will be increased, resulting in a lower drawbar pull in cold weather. One way of correcting for this is to add to the resistance of the engine truck, trailer and tender the same percentage as for freight cars, for cold weather. With the locomotive 0.8 of 22½ lb. per ton, the friction of driving wheels, rods, etc., or 18 lb. per ton, will be affected by low temperature, and this value should be corrected in the same ratio as the correction for car resistance.

An approximate method of correcting drawbar pull for temperatures of 50°F. and below is to deduct $\frac{1}{2000}$ of the drawbar pull at 70° and 5 m.p.h. for each degree that the temperature is below 70°F.

Frictional Resistance.—The term "Speed Resistance" is frequently used to cover this division of the subject, since the many frictional resistances noted earlier in this chapter vary with some function of the speed. This problem has been one in which engineers find much of interest and the printed discussions are extensive.[1]

Under average conditions, frictional resistance is found to range from 14 to 22 lb. per ton in starting single cars with an average of 18 lb. The resistance drops rapidly, and after reaching a speed of 6 to 8 m.p.h. it again increases as the speed increases but decreases as the gross weight of a car and



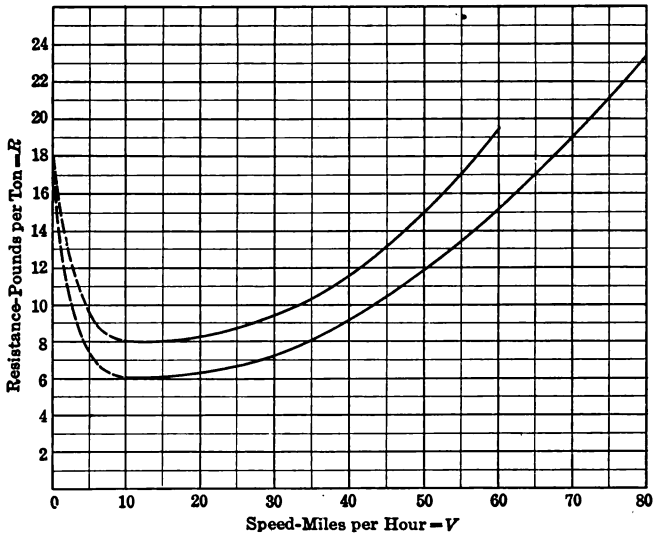
This curve is based on the formula $R = 4.3 + 0.0017\ V^2$, and should be used for cars weighing 45 tons and upwards. For lighter cars, use curves for freight cars of corresponding weights.

FIG. 22.—Resistance of passenger cars in pounds per ton at various speeds —Baldwin Locomotive Works.

its lading increases. Furthermore, the friction increases with decrease of air temperature. A complete analysis of the conditions includes a study of journal friction, but analysis based on theory alone fails to give reliable results. The working values to-day for train resistance as a whole are based on dynamometer car records and the curves commonly used are given in Figs. 22, 23 and 24.

Results of speed resistance accepted to-day may be discarded to-morrow. The curves, Figs. 22 and 23, are presented as safe workable values for passenger cars and for locomotive and tender, these being taken from "Locomotive Data," Baldwin

[1] This subject has been reviewed by Mr. F. J. Cole, Railroad Age Gazette, August 27 to Oct. 1, 1909.



Lower line applies to heavy standard gauge locomotives and tenders, and is based on formula $R = 4.3 + 0.0030\ V^2$. Upper line applies to narrow gauge and light standard gauge locomotives and tenders, and is based on formula $R = 5.0 + 0.0040\ V^2$.

FIG. 23.—Resistance of locomotives and tenders in pounds per ton at various speeds—Baldwin Locomotive Works.

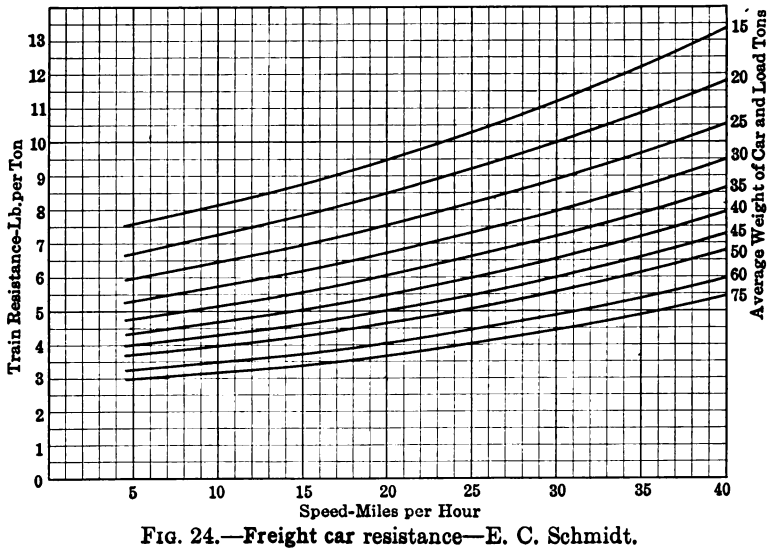


FIG. 24.—Freight car resistance—E. C. Schmidt.

Locomotive Works; and unless otherwise noted will be used in problems in this book.

Freight Car Resistance.—The Schmidt results shown in Fig. 24 are used widely. These are from extensive tests and have been reduced to the formula:

$$R = 1.5 + \frac{106 + 2V}{W+1} + 0.001\,V^2$$

W = Weight of car in tons.

V = Speed in miles per hour.

Certain roads have made an extensive study of car resistance, a comparison of some recent values being as follows:

COMPARISON OF RESULTS OF CAR RESISTANCE

From tests made by	C., B. & Q.	E. C. Schmidt	P. R. R.
Speed, m.p.h.	20	10	8 to 12
Weight per car, tons	Resistance per ton in pounds		
15	6.30	8.20	8.90
20	5.20	7.30	7.05
25	4.40	6.45	5.90
30	3.80	5.75	5.10
35	3.40	5.15	4.60
40	3.10	4.70	4.20
45	2.85	4.25	3.90
50	2.70	3.95	3.65
55	2.50	3.70	3.45
60	2.40	3.50	3.25
65	2.30	3.35	3.10
70	2.25	3.25	3.00
75	2.20	3.15	....
80	2.10	3.05	....
85	2.05	....	....
90	2.00	....	....

THE KIESEL TRAIN RESISTANCE FORMULAS

W = total weight of train in tons.
C = curvature in degrees.
G = grade in per cent.
N = number of cars.
V = speed in miles per hour.
R = total resistance in pounds.

The formula for resistance on a level tangent will be:

$$R = 100N + 1.5W + 0.01V(V + 16)\sqrt{WN} \qquad (42)$$

which is for the average train resistance, knowing that there is a possibility of obtaining results about 15 per cent. better when car, weather and track conditions are favorable. This does not include resistances due to grades and curves. The average for curve resistance is taken as 1 lb. per ton per degree, or CW.

Grade resistance equals 20 lb. per ton for each per cent. of grade, or 20 GW. With these terms added, the formula reads:

$$R = 100N + (1.5 + C + 20G)W + 0.01V(V + 16)\sqrt{WN} \qquad (43)$$

When the locomotive and tender are included in the train, the weight of train, W, in the above must include the weight of the locomotive and tender, and the number of cars N as used, is to be taken as the number of cars back of the tender plus three.

The foregoing arrangement enables one to handle the whole train weight, including that of the locomotive and tender, as a unit, not only for grades and curves, but also for resistance on straight level track.

The locomotive resistances consist of Machinery Friction and Head-end Wind Resistance, in addition to those mentioned in connection with train resistance.

For machinery friction assume,

$$R_1 = [22 + 0.15\ (n - 1)\ V]Q \qquad (44)$$

in which Q = weight on drivers in tons.
n = number of pairs of drivers.
V = speed in miles per hour.

For head end wind resistance use Professor Goss' Formula, Eq. 41,

$$R_2 = 0.1V^2$$

Where V = speed in miles per hour.

The locomotive and tender resistances, which are here taken the same as for the cars, are subject to the train resistance formulas and will be considered as part thereof, assuming that the engine and tender weights are equivalent to three cars of the same total weight as the engine and tender.

Locomotive capacity expressed in terms of drawbar pull behind the tender is the difference between the cylinder tractive power and all resistances.

Consider the available power of the locomotive as cylinder tractive force, less machinery friction and head end wind resistance,

$$A.T.F. = T.F. - R_1 - R_2$$

(For *T.F.* see Eq. 12, page 27)

or

$$A.T.F. = \frac{2PM}{1 + \frac{110w}{3} \times \frac{MV}{KH}} - [22 + 0.15(n - 1)V]Q - 0.1V^2 \qquad (45)$$

But, *A.T.F.* for limiting speed = *R*. Therefore,

$$\frac{2PM}{1 + \frac{110w}{3} \times \frac{MV}{KH}} - [22 + 0.15(n - 1)V]Q - 0.1V^2 = 100N + (1.5 + C + 20G)W + 0.01V(V + 16)\sqrt{WN} \qquad (46)$$

Problem 9.—Find the grade up which (*a*) a K2 and (*b*) a K2sa locomotive can each haul twelve passenger cars of 70 tons (total) each, at a uniform speed of 30 m.p.h. Steam superheated 200°F. in the K2sa. (Data for locomotives given in Table II, page 12.)

CHAPTER VI

NEW GRAPHICAL METHODS APPLIED TO LOCOMOTIVE PERFORMANCE

Characteristic Curves.—Referring to Fig. 25, let ABC represent the cylinder tractive effort of a locomotive for different speeds, this curve showing the maximum pull for a short distance. Draw MNO to represent at different speeds the total resistance of the train including locomotive and tender. Let the maximum speed of the train, which can be sustained over a considerable

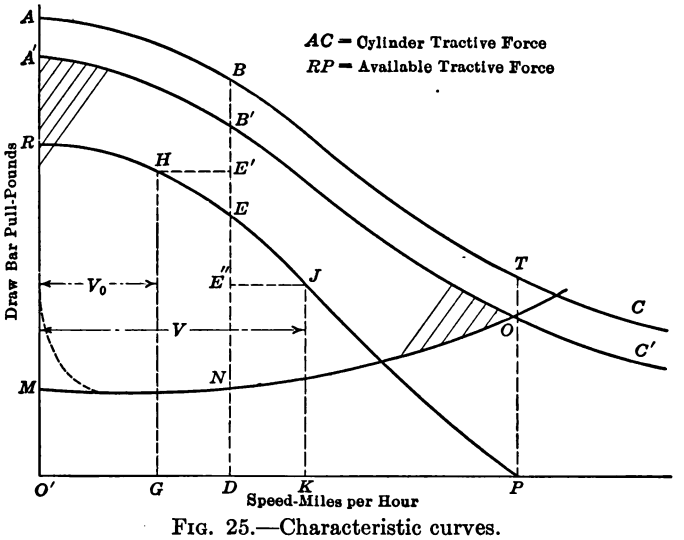


FIG. 25.—Characteristic curves.

distance, be at P (say at 50 m.p.h.) where the drawbar pull is PT and train resistance PO. The locomotive is capable of maintaining for a limited time, a drawbar pull greater, by the amount OT, than the resistance of the train on the level at that speed. This extra pull makes it possible to overcome abnormal resistances and to make up time. In order to reserve this pull at all speeds, replace the curve ABC by a curve $A'B'C'$ parallel to it and at a distance OT from the former. The shaded part of the diagram

limited by this curve and by the curve *MNO* represents the available power of the locomotive after satisfying all the requirements of the resistances when running on the level and at uniform speed and still keeping in reserve the pull *OT* to meet contingencies.

By plotting the speeds as abscissæ and the ordinates (as *NB'*) contained in the shaded area as new ordinates we obtain a curve *REP* which may be termed the "available," or better, the "unbalanced" tractive effort curve. Thus, to determine point *E* of this curve, make $ED = NB'$.

Assume that a train which has attained a speed *O'D* meets a 1 per cent. up-grade and that the extra pull for running up this grade is *F'*. Draw *DEB* and let some distance $DE'' = F'$.

One of three conditions may occur:

1. The point *E''* falls below the curve *REP*; in this case the locomotive can increase its speed and notwithstanding the grade attain a speed *O'K*.

2. The point *E''* falls at *E* on the curve *REP*; the locomotive can ascend the grade without loss of speed.

3. The point *E''* falls at *E'* above the curve *REP*; the locomotive will run more slowly and the speed will reduce to *O'G*.

Suppose it is required to find the time, *t*, and the distance, *S*, to increase the speed of the train from *Vo* (*O'G*) to a speed *V* (*O'K*). Assume that the part *HJ* of the curve is a straight line and that the mean drawbar pull on the level between points *H* and *J* is

$$DE = \frac{GH + JK}{2}$$

the time *t* in seconds and the distance *S* in feet are given by

$$t = \frac{2000W}{g} \times \frac{V - Vo}{DE} \tag{47}$$

$$S = \frac{2000W}{g} \times \frac{V^2 - V_o^2}{2DE} \tag{48}$$

these expressions being derived directly from the general equation, $F\ (\text{pull}) = \frac{W}{g} \times \frac{V}{t}$ where *W* is the weight in tons and *V* and *t* as given above.

For an up-grade of 1 per cent., the available drawbar pull will be reduced by an amount, say *F'*. By taking $EE'' = F'$ on the ordinate *DE*, the value of *DE''* will replace *DE* in the above equations.

For a down-grade of 1 per cent., DE will be increased by an amount F'. By taking $EE' = F'$ on the ordinate DE, the value of DE' will replace DE in the above equations.

PREDETERMINATION OF LOCOMOTIVE PERFORMANCE

The application of the "speed-time" curves in electric railroading has been thoroughly developed by C. O. Mailloux,[1] A. H. Armstrong and other engineers. The method in outline the same as used for electric railroads has been applied to practical cases of steam railroad operation, giving consistent results. The discussion here presented was first published by the author in the Railway Age Gazette (Mechanical Edition) Sept., 1914. Attention may also be called to a study of characteristic curves presented by Prof. W. E. Dalby before the Institution of Mechanical Engineers in October, 1912, and published in Engineering, November 1, 1912.

The problem before us is to find by graphical methods the least time in which a locomotive can haul a train of known weight over a given roadbed. This may readily be solved knowing a few leading dimensions of the locomotive, the weights moved and the grades and curves over which the run is to be made.

At the outset, we find that the problem for steam railroads differs from electric traction in that the tractive force exerted on a train may be maintained at a nearly constant value by electric motors from start up to the normal running speed of the train. The conditions are quite different with a steam locomotive. From start up to 50 or 60 r.p.m. the tractive force exerted by a locomotive is practically constant, the boiler supplying steam without drop in pressure. As the speed increases, the cut-off must be reduced in order to maintain steam pressure, the result being a decrease in tractive effort. The controlling factors determining the curve of maximum tractive force in steam operation are the weight on the drivers and the maximum boiler power. The weight on drivers is the controlling factor in electric traction.

The method for solving the problem, in outline, is as follows:

1. Draw the tractive force and resistance curves, finding for level track and on grades the available force at different speeds behind the tender. These may properly be termed the "charac-

[1] Transactions American Institute Electrical Engineers, Vol. XIX, 1902.

teristic curves" of a locomotive and have been discussed early in this chapter.

2. From results in (1) draw for level and for grades a "reciprocal" curve, which is a curve for different speeds, derived from the fundamental dynamic relation between the force F, acting on a mass M, producing an acceleration a, that is

$$F = Ma \text{ or } \frac{1}{a} = \frac{M}{F} = \frac{W}{Fg} = \frac{W}{F \times 32.16}$$

where F, the force, and W, the weight acted upon, must be in the same units. This will be a curve drawn from the reciprocal of the acceleration for different speeds.

3. Since velocity, $v = at$, the time, $t = \frac{1}{a} \times v$ (for uniform increase in speed) represents an area, obtain the time to accelerate the train by finding the area in proper units under a limited part of the reciprocal curve.

4. Since $S = vt$, obtain distance S, traveled in time t, by finding the area under the time-speed curve.

5. By aid of a profile map of the road, lay off the time-speed and time-distance curves for the train considered, by aid of curves for level and grades in (3) and (4) above.

Speed-time curves not only show the speed attained at any given interval of time, but they also show the variations in speed occurring at various intervals of time. The slope of a speed-time curve at any time-point is an indication and a measure of the time rate of change of speed at the corresponding instant of time; and it shows whether the speed is constant, is increasing or decreasing. A horizontal speed-line indicates constant or uniform speed. An upward slope in the speed-line indicates increasing speed, or acceleration; a downward slope indicates decreasing speed or deceleration. These characteristics serve to distinguish the different kinds of speed-time curves.

A Problem Solved

Division of road considered	Huntingdon to Tyrone, Pa. (a distance of 19.7 miles).	
Locomotive used	Atlantic Type, E2d.	
Locomotive and tender, total weight		140 tons.
Train composed of seven steel cars	one 60′ steel baggage	55 tons.
	one 70′ steel mail	69 tons.
	five 70′ steel passenger	306 tons.
Total weight		570 tons.

Weight on the drivers	61 tons
Boiler pressure = P	205 lb. per sq. in.
Diameter of piston = d	20.5 in.
Diameter of drivers = D	80 in.
Stroke piston = s	26 in.
Total heating surface	2640 sq. ft.
Steam	Saturated.

To illustrate the method, calculations are shown for 0.25 per cent. grade and 30 m.p.h.

Tractive Force and Resistance Curves.—It is first necessary to draw the cylinder tractive force curve of the locomotive.

$$T.F. = \frac{0.85Pd^2s}{D} \times \text{speed factor}$$

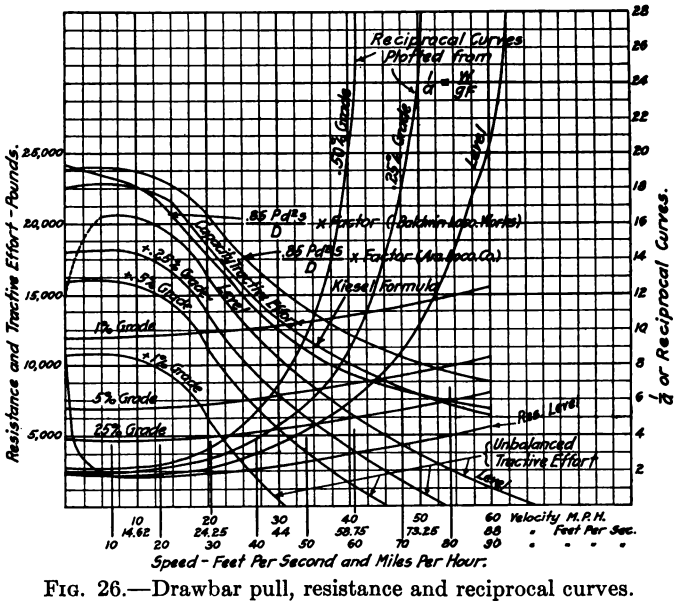


FIG. 26.—Drawbar pull, resistance and reciprocal curves.

where P is the boiler pressure; d, the cylinder diameter; s, the cylinder stroke, and D, the driver diameter, all in inches as given in Eq. 11, page 22. This may be applied, using speed factors of the American Locomotive Company or of the Baldwin Locomotive Works. To show how the results differ, both curves are drawn in Fig. 26. The author has preferred for this study to use the formulas developed by W. F. Kiesel, discussed on pages 27 and 58.

From Eq. 45, Kiesel's formula for the E2d class reduces to,

$$T.F. = \frac{53305}{1 + 0.0873V} - (22 + 0.15V)61 - 0.1V^2$$

where V is the velocity in miles per hour, and $T.F.$ the available tractive force, which is drawbar pull behind the tender. The part to the right of the fraction includes locomotive internal friction and wind resistance.

The resistance, R, for the cars hauled was plotted from the following:

$$R = 100N + (1.5 + C + 20G)W + 0.01V(V + 16)\sqrt{WN}$$

the terms being the same as in Eq. 43, page 57.

Figure 26 shows resistance curves for level track and for three up-grades but the formula is equally applicable to down-grades.

Unbalanced Tractive Effort Curve.—The unbalanced tractive force or effort at any velocity will be the difference between tractive effort values taken from the tractive effort curve and the values of resistance taken from the resistance curve. Therefore, to lay off the unbalanced or available tractive effort curves at different speeds, step off the distance between the tractive effort curve and the resistance curve for the grade in question and lay off these distances from the base line. These distances represent for different speeds the force available to accelerate the train and to overcome resistances not already taken into account. Where this curve and the base line intersect (which will be vertically below the intersection of the tractive effort and the resistance curves) will give the balancing or limiting speed, that is, the highest possible speed which could be reached, if the train runs under the conditions represented by these curves.

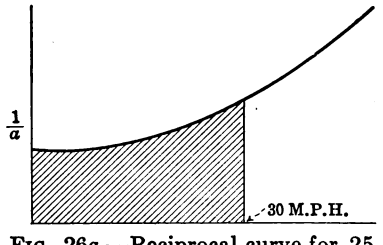


FIG. 26a.—Reciprocal curve for .25 per cent. grade.

Reciprocal Curve (Fig. 26).—Since,

$$F = Ma, \frac{1}{a} = \frac{W}{Fg} = \frac{W}{F \times 32.16}$$

Where F is the pull or force (in pounds) from the *unbalanced*

tractive effort curve, W, the weight in pounds of the train, including locomotive and tender, + 5 per cent. of the weight of train. This additional 5 per cent. takes care of the force required to overcome the inertia of the rotating parts during acceleration, and is an average value for widely varying conditions. The reciprocal $\frac{1}{a}$ of the acceleration is plotted for varying speeds differing by 10-mile units.

Time-speed Curves (Fig. 27).—The time-speed curve is plotted with velocities as ordinates and corresponding time elapsed as abscissas. The time corresponding to a certain velocity is found by taking the area under the reciprocal curve between two velocities, a convenient interval being a difference of 10 m.p.h.; it is therefore applying the general equation, $dt = \frac{1}{a} dv$. Suppose the plot of the reciprocal of the acceleration was made so that the

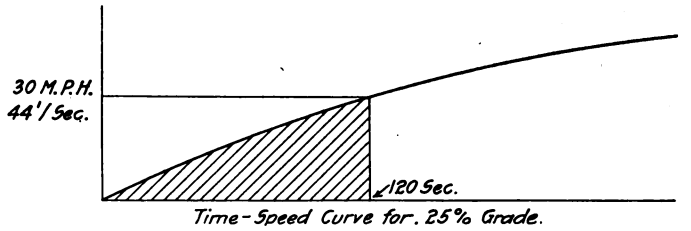


FIG. 26*b*.

larger of the unit squares into which the cross-section sheet is divided, equaled 2.0 (vertical scale) and the horizontal unit is 5 m.p.h., or 7.31 ft. per sec., as in Fig. 26. Then will the area of any one of these squares equal $2 \times 7.31 = 14.62$ sec. If a planimeter is used to determine the areas, the value of 1 sq. in. in the "seconds" units may be obtained in the same general way as when a large square on a cross-section sheet is taken as the reference unit.

The area under the reciprocal curve for 0.25 per cent. grade in Fig. 26, up to 30 m.p.h., gives a time when reduced to the proper units (as explained above) of 120 sec., and this value is plotted as shown in the upper diagram Fig. 27 and in Fig. 26*b*.

Time-distance Curves (Fig. 27).—Remembering that $dS = vdt$, or in general for uniform acceleration $S = vt$, the point corresponding to the space S passed over for $v = 30$ m.p.h. or (44 ft. per sec.) on the 0.25 per cent. grade, may now be determined by

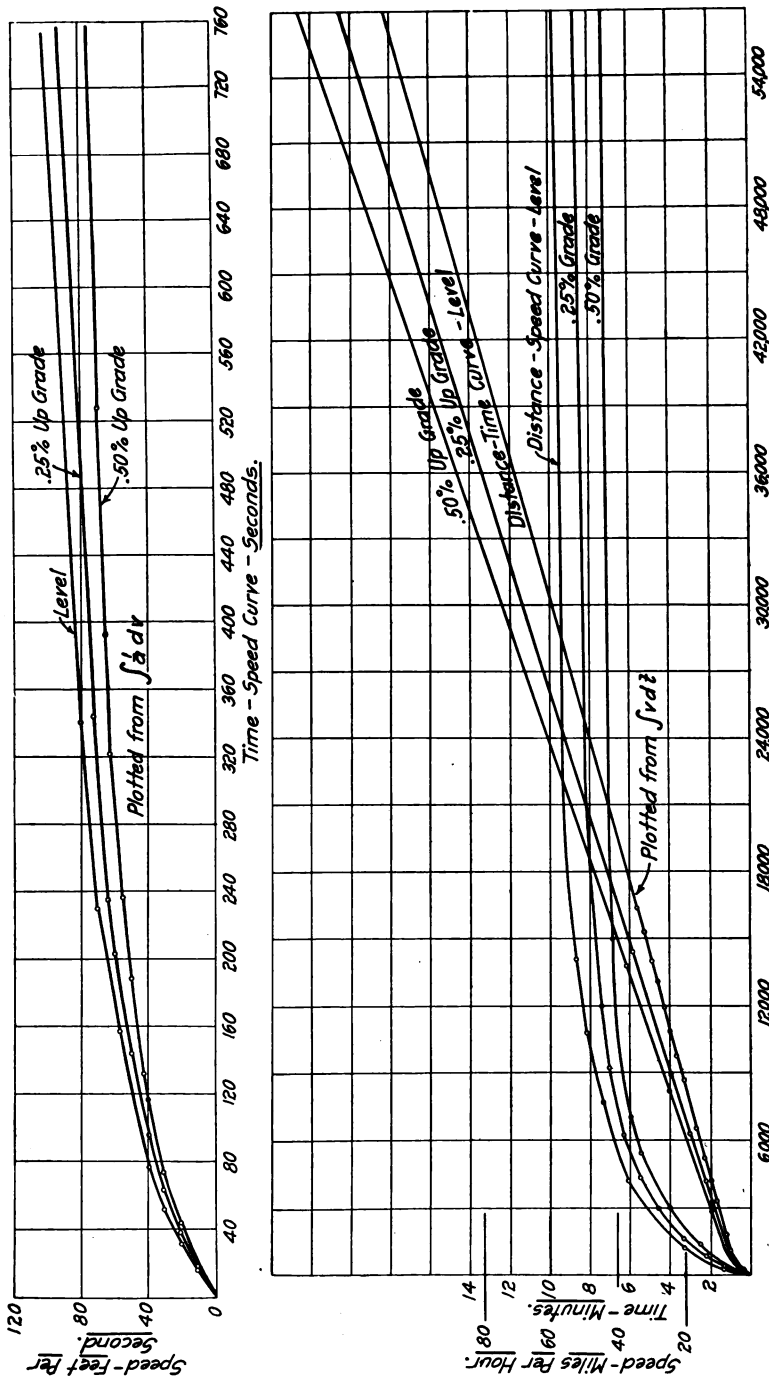


Fig. 27.—Time-Speed, Time-Distance and Speed-Distance Curves.

reducing the area (to the proper scale) up to 120 sec. on the time-speed curve. It is found to equal 3360 ft. This is the distance run while accelerating to 30 m.p.h. (See also Fig. 27*a*.)

Speed-distance (Fig. 27).—The speed-distance curve is drawn from results obtained by above method of plotting velocities as ordinates and distances corresponding to these velocities as abscissæ. The accelerating force will continue to act, increasing the speed until the limiting or balancing speed has been reached or nearly reached. However, the controlling speed over a division may not permit of as high a speed as is possible to attain, and in such case the locomotive will not be hauling to its limiting capacity. A correction may be made in such cases.

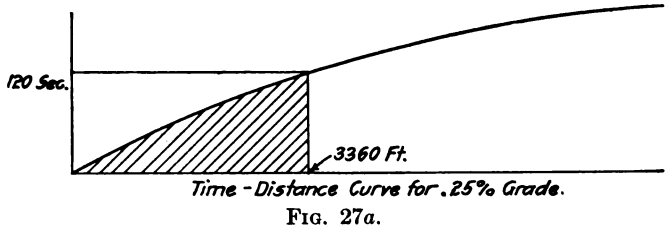


FIG. 27*a*.

Profile Compensated (Fig. 28).—The profile of the grade over which the train is to run is plotted on tracing cloth using the same distance scale as was used in the time-distance and speed-distance curves. The curves must first be compensated; that is, the grade reduced at the curve by such an amount that the total train resistance due to grade and curve will not exceed the maximum grade on a tangent. Each degree of curvature has been compensated to an allowance of 0.035 per cent. in grade for each degree of curvature. (See page 53.)

Combining Time-distance and Compensated Profile Curves (Fig. 28).—The tracing cloth on which is traced the compensated profile is placed over the distance-time curve in Fig. 27 so that the intersection of the co-ordinates at zero and the start of the grade coincide. The time curve is now drawn as far as the grade remains the same, interpolating between the curves to get the grade required. The tracing cloth is then moved *horizontally* until the end of the curve just drawn falls on the part of the time-distance curve corresponding to the new grade. Continue thus for the entire run and the final point on the curve will give the total time required to go the entire distance while the time to go any

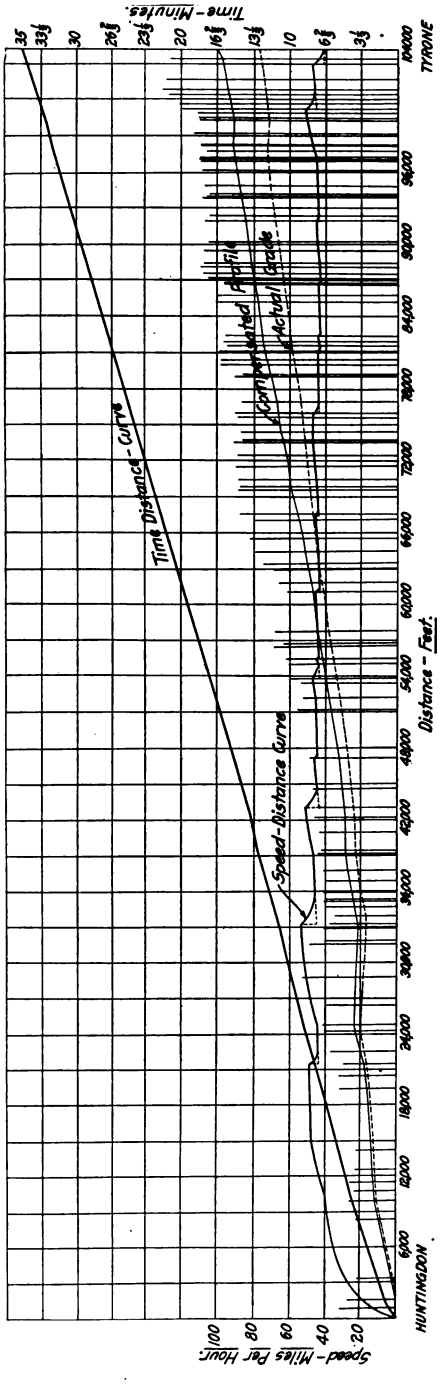


FIG. 28.—Time schedule for actual case.

distance from one point to another may readily be obtained from the curve.

Combining Speed-distance and Profile Curves (Fig. 28).—The tracing cloth is again placed so that the intersection of the co-ordinates at zero is made to coincide with the starting point on the profile and the speed-distance curve is traced off for the given grade, interpolating between the grade curves drawn in order to approximate to the proper grade curve. This curve is traced as far as that grade remains the same, the tracing cloth is then moved *vertically* until the end of the line traced coincides with the new grade curve and this curve is then traced.

The speed-distance curve will often fall very rapidly and even abruptly due to the fact that a train ascending a grade steeper than the one on which it has been running will slow down. It is evident that it will take a certain distance to adjust its speed to

the new grade conditions. To determine this distance Eq. 20*a* may be applied.

$$S = 70 \frac{V_2^2 - V_1^2}{F_t}$$

where S is the distance run (in feet) to change from higher velocity V_2 (before approaching the steeper grade) to velocity V_1, these velocities being in miles per hour. F_t is the tractive force in pounds per ton obtained from the unbalanced tractive force curve for the grade considered, and for the average of the two speeds V_2 and V_1. Another method is to apply Eq. 34, page 43.

For the analysis, here discussed, a section of track was chosen in which there are practically no down-grades or level track. The method applies equally well to such cases and also to stopping the train by application of the brakes.

Speed-time curves may be drawn as in Fig. 29, which shows on one diagram the results for level-track conditions replotted from Figs. 27 and 28. This composite view of the curves enables the reader to follow and read the results from quadrant to quadrant. Thus for 40 m.p.h. we read directly that it requires 178 sec. for the train in question to be accelerated from zero to 40 m.p.h., on a level track and further that 7500 ft. were passed over during the acceleration, the energy consumed being equal to 53,000 ft.-tons, the last result being from the curve added to the former plots thus extending the applications of this unique method. The final value above gives the foot-tons consumed by the train in accelerating from rest to 40 m.p.h. on a straight level track. For 36.5 m.p.h., follow dotted line *A–B–C–D*.

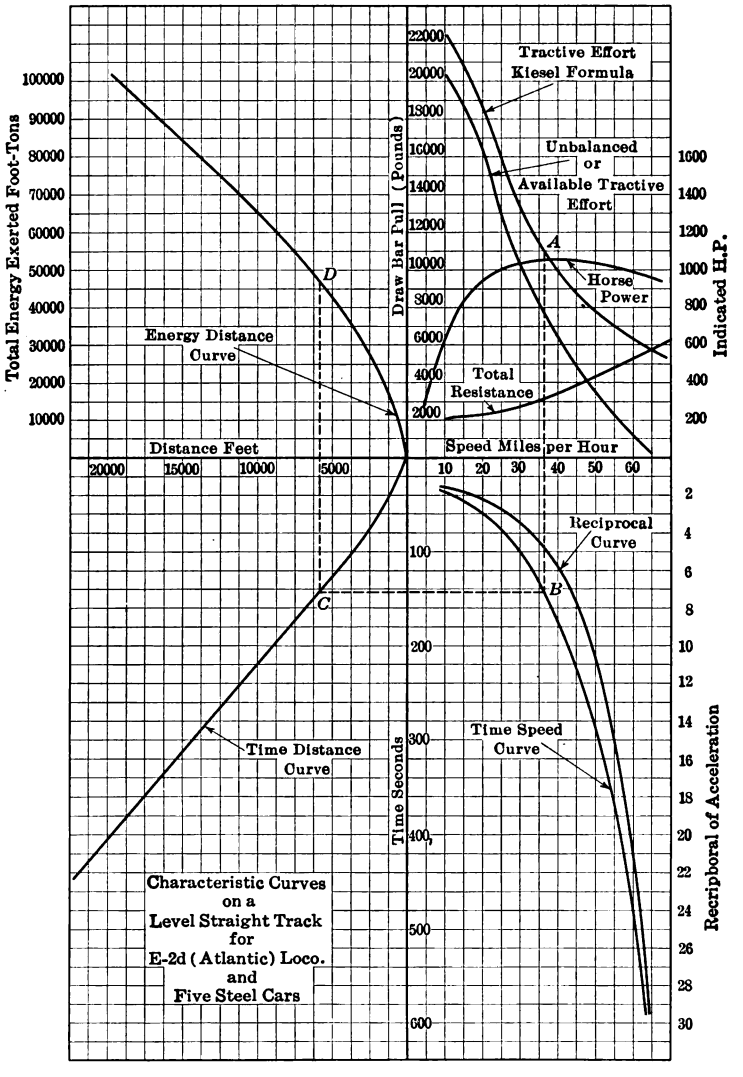


FIG. 29.—Plot for level track; speed-time curves.

CHAPTER VII

DYNAMOMETER CAR TESTS AND TONNAGE RATING

DYNAMOMETER AND TRACK INDICATOR CARS; USES OF TEST CARS; CORRECTIONS FOR ACCELERATION AND GRADES; EXAMPLE WORKED BY ADJUSTED TONNAGE METHOD

A dynamometer is an instrument for measuring and recording the force exerted and the work performed by machines. When such a device is mounted on a car and arranged to record the pull or push of the engine or the train resistances we have a dyna-

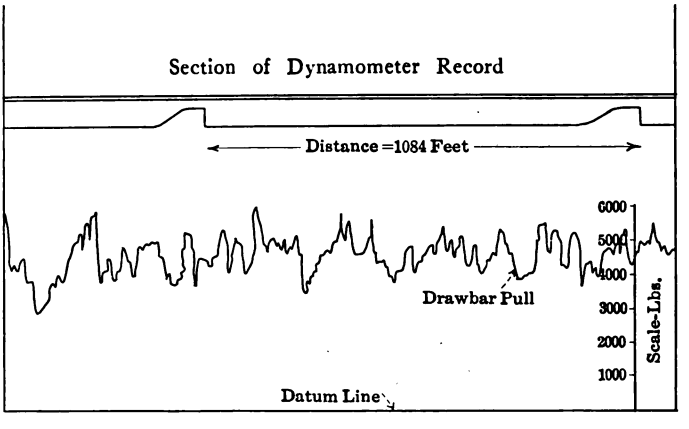


FIG. 30.—Record from simplest form of track dynamometer.

mometer car. The simplest form of a dynamometer consists of (*a*) a spring whose deflection indicates the force exerted on the car, and (*b*) a roll of paper moving at a rate directly proportional to the distance traveled by the car.

The first dynamometer used by the author was constructed entirely by students and at a cost of not over $40. It was simple in design but sufficiently strong to properly take care of a drawbar pull up to 18,000 lb. Reference to this is made simply to show what may be done by an inexpensive device. A heavy short spring was obtained and after being carefully calibrated

between the compression heads in a testing machine, it was mounted so that the pull or push of the engine made a record on a roll of paper, above a datum or zero line, 4000 lb. being equal to 1 in. vertical rise of the pencil. The roll was driven from the axle of the tender at a speed directly proportional to the speed of the train and a distance curve was recorded, an offset

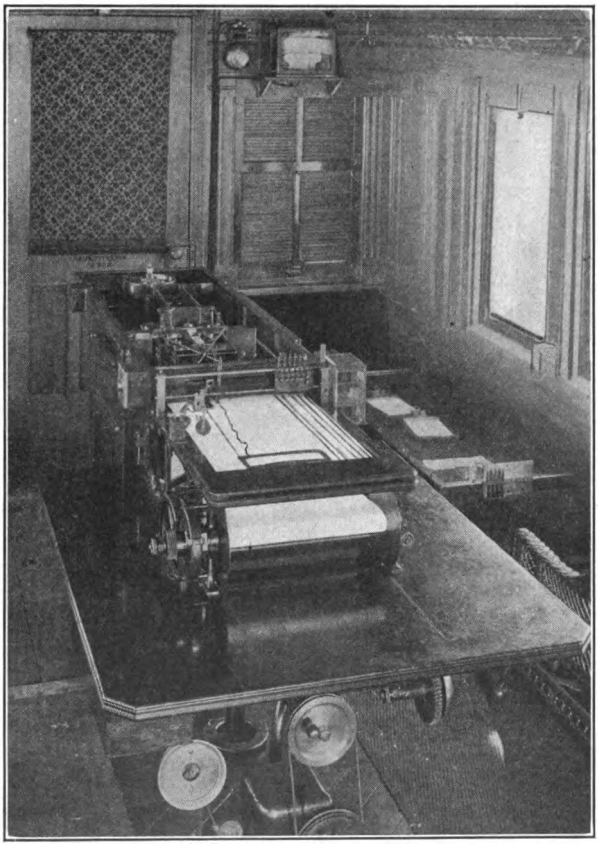


FIG. 31.—Interior view of Dynamometer Car, No. 100, owned by The Pennsylvania State College.

in which was made every 1084 ft. A record from a test made with this dynamometer and the college locomotive is shown in Fig. 30. This was taken over a poor stretch of track; but by drawing a mean line through the drawbar record, the average pull behind the tender can be obtained with fair accuracy. The

dynamometer horse power at any point may be determined and compared with the indicated or cylinder horse power, the difference between the two giving the internal frictional resistance expressed in horse-power units. The ratio of the dynamometer to the indicated horse power is the mechanical efficiency of the locomotive.

Thirty years ago, the Dudley "Dynagraph" proved the practical value of a car with apparatus equipped as here discussed. Later, the Pennsylvania Railroad Company built its first dynamometer car from designs of Mr. Charles E. Emery. This car was presented to The Pennsylvania State College in 1912, where it is used in road testing. An interior view of one end of the car, Fig. 31, brings out some of the principal features.

The dynamometer springs, which are also the draft springs, are under the flooring of the car, and the motion from compression of the springs is transmitted by levers to the recording pen.

The method of autographically recording the different observations is shown in the accompanying illustration. As the paper moves along at the rate of exactly 1 in. for every 100 ft. traveled by the car, a base line (which is the zero position of the pen recording the drawbar pull) is marked on the paper. As the pull increases, a stylographic pen moves at right angles above the datum or base line an amount proportional to the pull, the scale being 3990 lb. to the inch. Each square inch under such a recorded pull represents, therefore, $3990 \times 100 = 399{,}000$ ft.-lb. In the illustration, this pen is shown a little to the left of the center of the paper. The drawbar end of the car being placed next to the tender, the pull recorded includes the dynamometer car as part of the train behind the tender.

The horizontal arm, shown in the rear, balances the arm from which the record is made on the paper. Back of this horizontal arm may be seen a vertical rod supporting a wheel on a disc attached to the horizontal arm, which in turn acts (by means of a dash-pot) to prevent severe shocks on the weighing mechanism, when the load is applied at the drawbar. The five pens, shown at the right, give straight-line records, except when they are moved by magnets giving a slight offset when making some special record. The roll of paper may readily be removed and the record worked over at any convenient time or place.

In a later and much larger capacity car, also designed by Mr. Emery for the Pennsylvania Railroad, the load is carried on the

piston of a large hydraulic cylinder.[1] The pressure on the liquid is transmitted to a small cylinder, the piston movement of which is controlled by the compression of one or more springs. The

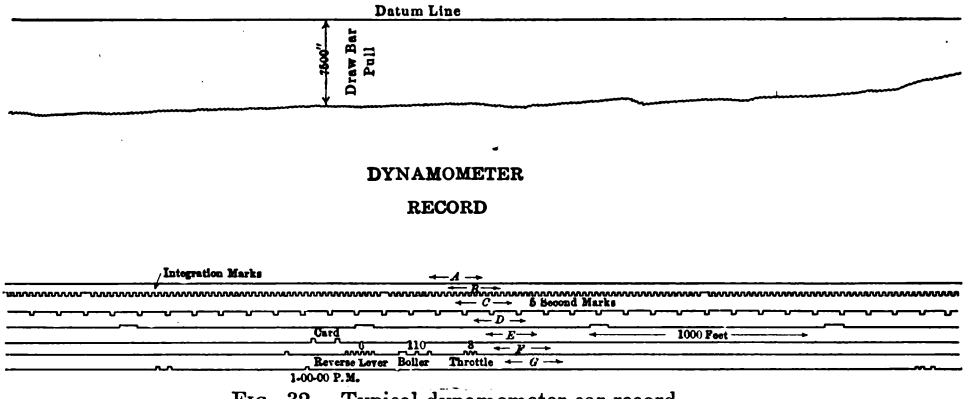


FIG. 32.—Typical dynamometer car record.

Line *A* shows whether load is "pull" or "push."
Line *B* shows the output, each mark representing 200,000 foot-pounds of work. The drawbar pull is continuously integrated.
Line *C* shows marks made at equal intervals of time, 5 seconds in this case.
Line *D* shows distance travelled. From mark to mark represents 1000 ft.
Line *E* shows when indicator cards are taken.
Line *F* is recorded from the cab and gives boiler pressure, throttle and reverse lever positions.
Line *G* shows mile posts, bridges, curves and towers.

amount of pull or push is recorded graphically (always on the same side of the datum line) by a pen fastened by means of an extension rod to the springs.

Test No. 10. Wt. of train, 534
No. cars, 8 No. wheels, 88

Section	Distance			Ave. D.B.P.			Ave. speed			Ave. resistance cor. for acc.		Rest. cor. for grade and acc.
	Total	Push	Pull	Actual	Corrected Acc.	Corrected Acc. + grade	Throttle open	In. motion	Total time	Per Car	Per ton	Per ton
				A	*B*	*C*	*D*	*E*	*F*			
1	62,281	831	61,450	22,600	21,770	4,440	30.18	30.20	30.20	2721.25	40.09	8.17
2	71,207	17,900	53,307	6,280	3,680	5,080	53.99	51.54	51.54	460.00	6.77	9.35
3	29,068	19,768	9,300	5,440	2,980	5,840	64.70	56.77	56.77	372.50	5.48	10.75
4	64,677	.....	64,677	6,680	6,220	4,120	50.34	50.34	50.34	777.50	11.45	7.58
5	27,125	21,875	5,250	7,150	6,200	8,530	55.92	56.54	56.54	775.00	11.41	15.70

NOTE: Abbreviations in table as follows:
Ave. = average. Cor. = corrected.
Acc. = acceleration. Rest. = resistance

[1] American Machinist, May 30, 1907.

Seven electrically worked pens give records shown in Fig. 32.

In working up results, a form similar to the foregoing is filled in. The values here given as an illustration, were taken from a test to determine car resistance.

There are many uses to which dynamometer cars are applied. An illustration of one special application is brought out in Fig. 18, Chap. IV. By a record as there presented, we may study all the leading characteristics of train acceleration including locomotive capacity and may separate acceleration, grade, speed and other resistances into their component parts, examining the chief characteristics of each as related to that particular test.

Track Indicator Cars.—All dynamometer cars are built along the lines indicated in the foregoing brief descriptions. They should not be confused with track indicator cars, designed to ascertain track conditions and made chiefly to safeguard high-speed trains and further for the purpose of comparison with established track standards. In such cars, the apparatus for the detection of track irregularities is designed to automatically indicate, often by the ejection of colored liquids upon the road-bed, imperfections of track alignment and to make autographic records of the following conditions: (1) Irregularities of track surface. (2) Variations of gage. (3) Cross level and car swing. (4) Lurches. (5) Speed and distance traveled. (6) Time in 5-sec. intervals.

This apparatus provides means for the detection of low joints and unsatisfactory conditions of track surface and as sometimes designed, vertical movements of the central axle of a six-wheel truck relative to the outer axles are made to indicate the existence and extent of imperfect surface alignment.

Fluctuations in track condition are recorded upon a strip of moving paper, by means of stylographic pens. These pens are arranged in two sets, one of which being stationary, establishes a series of horizontal datum lines, the other pens have a motion at right angles to the datum, which motion is controlled by the various track imperfection detecting mechanisms and is limited by the extent of these imperfections. As the paper moves under the pens at a rate directly proportional to the car speed, indications of track irregularities will appear in their true relation and can be readily located.

TONNAGE RATING

Dynamometer cars are used chiefly to rate locomotives over a division. Rating consists in determining for a given locomotive the tonnage it can haul over a given roadbed; *it is the balancing of an equation between the power of a locomotive and the resistances of the train, under the limitations imposed by a particular roadbed and some allowable speed.*

In Chap. V were noted the conditions affecting train resistance and therefore influencing the amount of tonnage which can be hauled by a particular locomotive. In addition to the conditions mentioned in working up any tonnage rating system, the following operating conditions must be considered:[1]

1. Speed of train on ruling grades; with increase of speed, locomotive drawbar pull will decrease, while car resistance will increase as heretofore discussed.
2. Average speed desired between terminals.
3. Capacity of fireman or mechanical stoker.
4. Condition of fire affected by number of delays.
5. Number of stops.
6. Location of block signals with reference to ruling grade.
7. Location of sidings and other stopping points with reference to ruling grade.
8. Grade of track at sidings.
9. Location of momentum grades.

When using a dynamometer car for tonnage rating, the accuracy of the results depends primarily upon the true calibration of the dynamometer springs. The author has used the following simple method found to be accurate and easily applied: A bar of steel of proper cross-section is calibrated in tension by a testing machine, using an accurate and close-reading extensometer. Without disturbing the extensometer on the test bar, it is applied to the springs of the dynamometer through the medium of a straight steel bar, which displaces the drawbar of the coupler. Applying the load by means of hydraulic jacks, the bar is extended as it was in the testing machine. After taking the readings from the car, the bar is re-calibrated in the testing machine.

[1] From report of Committee on Train Resistance and Tonnage Rating, A. R. M. M. A., June, 1914.

The methods of train loading in common use, are:

A. Drawbar pull method.
B. Adjusted tonnage method.

If speed resistance was the same for an empty as for a loaded freight car, the problem would be simple, for the entire weight of train in tons times the resistance per ton would give the pull required behind the tender.

A. To determine the train loading by the drawbar pull method, it is first necessary to know the weight of each car, and from dynamometer car records (or from resistance curves for different weights of cars, as shown in Fig. 24) find the resistance of each car. A profile of the road is then examined, from which the ruling grade is determined and the resistance of each of these cars as found above must be added to the resistance due to grade alone. From these new values for the total resistance on the ruling grade a table is prepared giving the resistance for each weight of car. When the sum of the resistances of the individual cars is equal to the drawbar pull, the locomotive is loaded. *Therefore, this method consists in finding the resistances to overcome and equating these to the available power of the locomotive.*

This is the most accurate of any method, but is inconvenient to apply.

B. The above has been simplified by replacing the actual gross tonnage by an *adjusted tonnage,* to which all trains are loaded. This adjusted tonnage is such that the resistance or drawbar pull per adjusted ton is the same for all trains to be hauled by a locomotive of a given class, regardless of the individual weight of the cars making up the train. This gives a tonnage figure, or load, the same for each train on a division.

To obtain this adjusted tonnage, a quantity C is added to the gross weight of each car and this quantity is called the adjustment or car factor.

Let:

Wl = total weight of loaded or heavy cars in first train.
We = total weight of empty or light cars in second train.
Nl = number of loaded cars.
Ne = number of empty cars.
C = car allowance or adjustment factor.

$$C = \frac{Wl - We}{Ne - Nl} \tag{50}$$

Adjusted tonnage loaded cars = $Wl + Nl \times C$.
Adjusted tonnage empty cars = $We + Ne \times C$.

EXAMPLE

Assume a consolidation locomotive weighing, with tender, 156 tons. Weight on truck wheels 10.5 tons, weight on drivers 86.6 tons, weight of tender 58.9 tons. Empty cars weigh 20 tons and loaded cars, 72 tons.

To calculate tonnage handled on a 0.5 per cent. grade at 70°F. when moving at 10 m.p.h.

At 70°F. there will be no reduction in drawbar pull due to low temperature.

Drawbar pull at 70°F. and 10 m.p.h. is found to be 33,200 lb.

Resistance of locomotive and tender due to grade, 156 × 0.5 × 20 lb., 1560 lb.

Leaving available for hauling train, 31,640 lb.

From Fig. 24, page 55.

Rfe = frictional resistance of empty cars, 7.3 lb. per ton.
Rg = resistance due to grade 0.5 per cent. 10 lb. per ton.
Re = total resistance of empty car, 17.3 lb. per ton.
and
Rfl = frictional resistance of loaded cars, 3.2 lb. per ton.
Rg = resistance due to grade 0.5 per cent., 10.0 lb. per ton.
Rl = total resistance of loaded cars, 13.2 lb. per ton.

We = weight of train empty cars $= \frac{31{,}640}{17.3} = 1835$ tons.

Ne = number of empty cars in above train $= \frac{1835}{20} = 91.75$ cars.

Wl = weight of train loaded cars $= \frac{31{,}640}{13.2} = 2400$ tons.

Nl = number of loaded cars in above train $= \frac{2400}{72} = 33.3$ cars.

C = car allowance factor (by Eq. 50).

$$\frac{2400 - 1835}{91.75 - 33.3} = \frac{565}{58.45} = 9.7 \text{ tons per car}$$

Train loaded cars, 2400 + 9.7 × 33.3 = 1723 adjusted tons.
Train empty cars, 1835 + 9.7 × 91.75 = 1725 adjusted tons.

The adjusted tons for different temperatures may be calculated in the same manner as given above, using the values for train resistance as affected by temperature conditions noted on page 53, Chap. V.

Problem 10.—(*a*) Calculate the adjusted tons for problem worked above, but for 5 m.p.h. and for $-10°$ and $+10°$. (*b*) Calculate the adjusted tons for conditions in above problem, except that grade = 0.4 per cent.

Equivalent Grade.—It is found in many cases that a ruling grade can be approached at a fairly high speed, say from 25 to 45 m.p.h. Where this initial speed can be depended upon, it will be possible to haul heavier trains than those calculated for a constant speed or dead pull.

One formula used in such a calculation is:

$$G \text{ per cent.} = 3.5 \times \frac{V_2^2 - V_1^2}{S} \qquad (49)$$

where G per cent. = per cent. of grade to be deducted from actual grade.
V_2 = initial speed in miles per hour.
V_1 = speed at top of grade in miles per hour.
S = length of grade in feet.

Having found G per cent. as above, the equivalent grade is found by deducting G per cent. from the actual grade, and tonnage rating is calculated for the equivalent grade, as previously outlined.

Physical relations of rise in feet to length of grade in connection with possible initial speed will have to be considered when determining whether or not a grade can be handled in this manner. Other considerations in this connection are the location of stations, towers, water tanks, sidings and block signals, which, if placed on the grade, will prevent its being considered as a velocity grade.

Eq. 49 is the equivalent of Eq. 20*a*, page 35, and is derived as follows:

From Eq. 20*a*, $\qquad F_t = 70\frac{V_2 - V_1}{S}$

Since 1 per cent. of grade offers a resistance of 20 lb. per ton, the value of G per cent. is obtained by dividing above by 20, deducing directly Eq. 49.

QUESTIONS ON CHAPTERS IV, V, VI, AND VII

1. Explain by reference to Fig. 18 what is understood by "uniform" and "variable" acceleration.

2. How does down-grade affect the determined points in the "corrected drawbar pull" curve, Fig. 18. Read from this same diagram what space, in feet, was passed over in accelerating up to 32 m.p.h. What space up to 45 ft. per sec.?

3. Define "virtual" grade and explain its application by reference to Fig. 28.

4. Why is the resistance in starting so much more than when under speed? How could the starting resistance be appreciably reduced?

5. Why are not the curves in Fig. 16, page 36, straight lines if the rate of change of velocity is uniform?

6. Is grade resistance influenced by speed? Give reason for your answer.

7. If the outer rail is properly elevated on a curve for a car running at 60 m.p.h., what will be the effect if speed is 40 m.p.h., while rounding the curve?

8. Why should the number of pairs of drivers influence machinery friction? (See Eq. 44.)

9. Referring to Fig. 29, review each step in the application of speed-time curves to locomotive operation, bringing out the importance of the use of the "reciprocal" curve.

10. How is the correction properly made on speed-distance curves, when a train approaches a grade steeper than the one on which it has been running?

11. Define "adjusted tonnage," "locomotive rating." Describe method of calibrating dynamometer springs; method of correcting for acceleration and for determining the equivalent grade.

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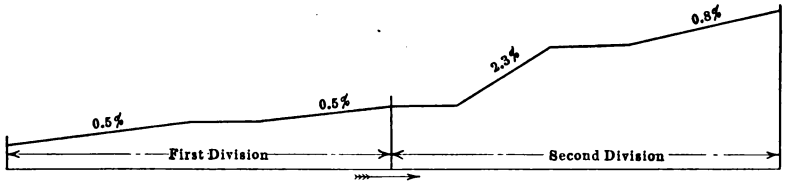
10. Special Review Problems.—(*a*) Select a locomotive with sufficient capacity for handling fast freight trains weighing 1600 tons, over the line



whose profile is shown above. On the first division, the minimum speed is to be 20 m.p.h.; while on the second division, a speed of 10 m.p.h. is to be maintained on the 2.3 per cent. grade, and a speed of 15 m.p.h. on the 0.8 per cent. grade. The locomotives used must not carry more than 50,000 lb. on each pair of driving wheels.

(*b*) By the application of speed-time curves, Chap. VI, find (1) the time to run over the first 0.5 per cent. grade in First Division if this grade is 5 miles in length. Locomotive and train (as in Chap. VI) starts from rest at bottom of grade. (2) How much shorter time is required if it "hit the grade" at 25 m.p.h.?

CHAPTER VIII

AIR BRAKES

DEVELOPMENT; INTRODUCTION OF TRIPLE VALVE; SIMPLE EXPLANATION OF AUTOMATIC BRAKE; LIMITATIONS OF EARLY TYPES; PRINCIPLES OF BRAKING DEVELOPED; METHOD OF DETERMINING LENGTH OF STOP; BRAKING RATIO; RESULTS OF BRAKE TESTS; BRAKE RIGGING; BRAKE DESIGN; PENNSYLVANIA-WESTINGHOUSE TESTS; STRESSES BETWEEN CARS DURING BRAKING

The history of braking reaches back some three centuries, but the art of braking by air pressure covers a period of less than 50 years. In 1833, Stephenson secured patents on a brake in which steam acted on a movable piston and the retarding force was applied through a system of rods to the brake shoes. In 1844, Nasmyth and May brought out the first pneumatic brake and which was worked on a vacuum. Four years later Lister patented an air brake having an axle-driven compressor and with general features similar to those of the straight air-brake system of to-day.

The modern air brake dates from 1869, when George Westinghouse invented what is commonly known as the Straight Air Brake. This consisted essentially of a steam-driven air compressor placed on the engine, a reservoir in which the compressed air was stored, a pipe line extending throughout the length of the train with flexible hose and coupling between the cars and a cast-iron brake cylinder on each car, the piston rod of which was connected to the brake rigging in such a way that when compressed air was admitted to the cylinder the piston was forced out and the brakes thereby applied. By means of a three-way cock placed in the cab of the locomotive, air could be admitted to the brake pipe and thence to the cylinder on each car, thus directly applying the brakes; or the air could be discharged from cylinder and train line, thus releasing the brakes.

Straight air was found to be dangerous for train service on account of chances of a hose breaking, resulting in entire loss of

braking force. Therefore, it was necesssary to have a truly automatic system and this led to the introduction by Westinghouse in 1872 of the automatic brake. The essential difference between the automatic and straight air consists in reversing the pressure conditions in the piping; in the automatic the brakes are applied when the pressure is lowered in the brake pipe and are released when this pressure is raised. In connection with this system, Westinghouse brought out the TRIPLE VALVE, the most significant and far-reaching invention in the development of air brakes.

As the name implies, the triple valve has three functions: (1) Charging the auxiliary reservoir, (2) applying the brakes, and (3) releasing the brakes. By means of the engineer's brake valve in the cab, it became possible to accurately control the intensity of application of the brakes but the engineman had no control over the graduation of release of the brakes, the brake releasing locally on each car. The brake was applied from air pressure in the auxiliary reservoir, by a *decrease* in the pressure in the brake pipe line, the connection being made to the brake cylinder through the triple valve. The brake thus became truly automatic in its action, for it could be applied from the cab or it applied itself in case of excessive leakage, from a bursted air hose, or for any cause resulting in a material drop in brake pipe pressure.

The action of the automatic brake may be explained by aid of the sketches, Figs. 33 and 33*a*, which are merely diagrammatic. Air is compressed by a steam-driven air compressor (placed on the engine) which is automatically stopped by means of a governor when the desired air pressure is reached. The compressed air is delivered to main reservoirs, which in turn connect through the engineer's brake valve (placed in the cab) with the brake pipe. This brake valve corresponds to a three-way cock *C* having functions shown in the illustrations.

Under each car is a triple valve, an auxiliary reservoir and a brake cylinder, the triple valve controlling air to and from the brake cylinder. Fig. 33 shows the condition when brakes are released. When the engine is coupled to the train, air is admitted to the brake pipe and passing into the triple valve acts on a piston which, in connection with a slide valve, controls the passage of air through ports, so that the auxiliary reservoir is charged from the brake pipe. At the same time, air is released from the brake cylinder as shown.

If the brake pipe pressure is reduced by moving the brake valve handle as in Fig. 33*a* the triple valve is so moved that air is admitted directly from the auxiliary reservoir into the brake cylinder. By again raising the brake pipe pressure to exceed that of the auxiliary reservoir, the valve again assumes the position of brake release, Fig. 33. The force with which the brakes are applied depends upon the reduction of brake pipe pressure, a slow reduction causing the air to pass from *B* to *D*, thus building up gradually the pressure in *D*.

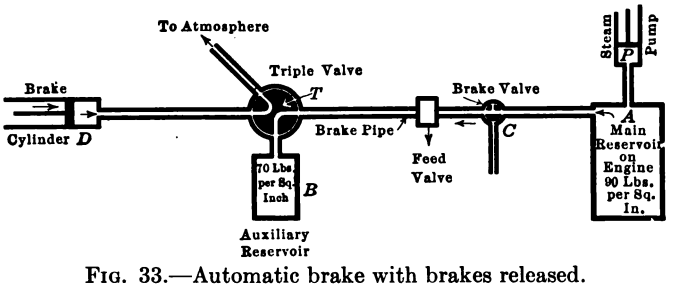


FIG. 33.—Automatic brake with brakes released.

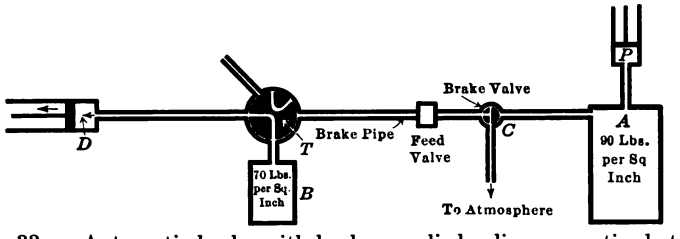


FIG. 33*a*.—Automatic brake with brakes applied—diagrammatic sketch.

In emergency application, air pressure in the brake pipe is quickly reduced and train-line air rushes directly into *D*. This reduction takes place in the engineer's brake valve or may be caused by a break anywhere along the brake pipe line. When the brake pipe pressure is reduced to a pressure below that in *B*, the brake pipe is automatically cut off from the auxiliary reservoir.

The plain automatic brake answered the purpose for light cars and short trains, but when it was introduced in freight service on long trains, it was found that its action had to be increased in rapidity because the brakes would be fully applied on the front cars of the train before they began to apply on the rear cars, which resulted in the rear cars running forward and taking up the slack with such violence as to cause severe shocks.

To overcome the limitations of the brake just described, the Quick-action Automatic Brake was soon brought out, marking the second step in advance. This brake was designed to operate on every car of the train before the slack was taken up or "run in." It gave a serial quick action and a higher braking force in emergency stop. To this system was added in 1892 the third great development, that of the High-speed Reducing Valve. This device was designed to use a higher pressure in the system, limiting it in a service application of the brakes to what was regarded safe and necessary, but during emergency application to allow the brake cylinder pressure to rise considerably higher than the maximum allowable in service application and then to

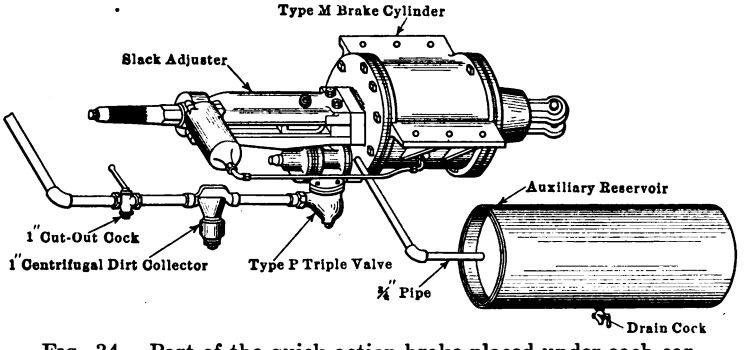


FIG. 34.—Part of the quick action brake placed under each car.

cause a gradual reduction of brake cylinder pressure, slow at first but more rapid as the train nears a stop, down to service limit. During emergency application, the force to retard the cars was increased some 50 per cent. when the brakes were first applied, which was gradually reduced until it became normal as it approached the end of a stop. Under each car is placed that part of the equipment shown in Fig. 34.

ET Equipment.—At first the type of brake equipment used on the engine and tender was practically the same as that used on the cars; however, experience required additional appliances and modifications of pressure controlling devices, in order to meet the various conditions of freight and passenger service, as well as flexibility in handling the locomotive alone. These additions resulted in a complication of apparatus which was difficult both to install and maintain. The present standard air-brake equipment is called the "ET," being an abbreviation of Engine and

Tender and is a combination of all previous equipments extremely simplified. Fig. 35 is a diagram showing how the equipment is

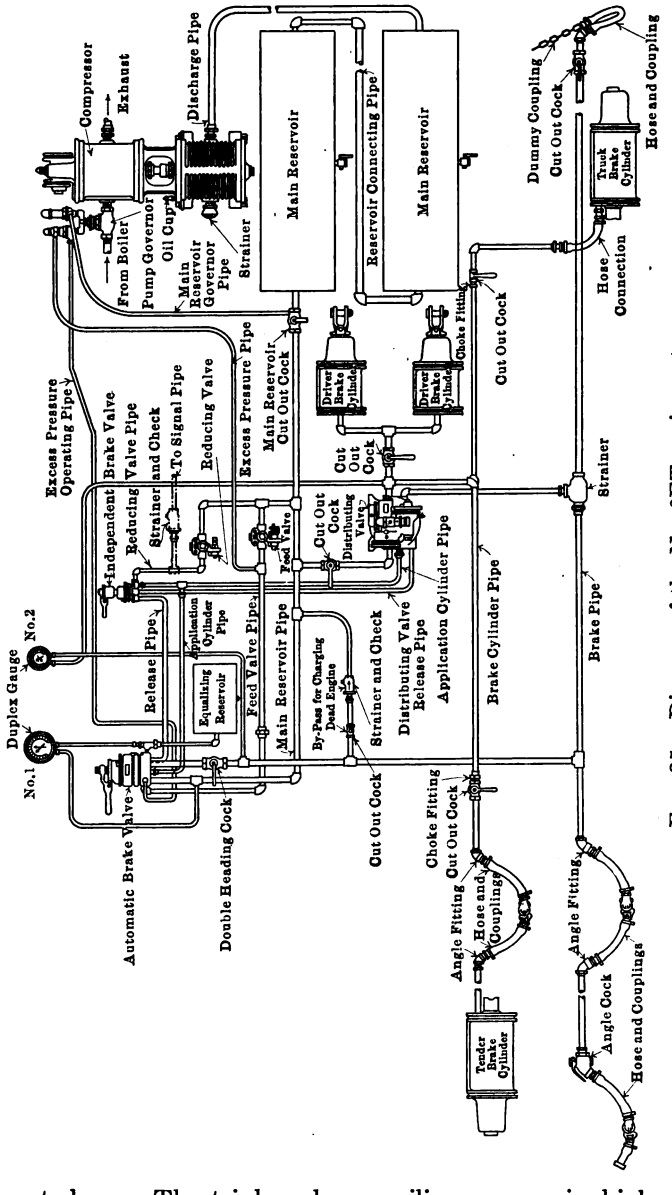


FIG. 35.—Diagram of the No. 6ET equipment.

connected up. The triple valve, auxiliary reservoir, high-speed reducing valve and some of the smaller parts, as used in the equip-

ment previously referred to, are combined into one device known as the Distributing Valve. It also requires an additional brake valve by means of which (in combination with the distributing valve) the brakes on the locomotive and tender may be operated independently of the brakes on the cars of the train while, at the same time, the usual automatic brake is available for operating the entire train.

Since the air brake was invented, that is, since 1870, non-articulated locomotives have increased in weight on drivers from 25,000 to 400,000 lb., in drawbar pull from 10,000 to over 100,000 lb., and in total weight from 90,000 to 750,000 lb. and the working steam pressures have increased from 125 to 225 lb. Freight cars have increased from 9000 lb. light weight to 48,000 lb., and in carrying capacity from 40,000 to 150,000 lb. The length of freight trains has increased from 15 to 130 cars and tonnage from 300 to 4500 tons. Passenger equipment cars have increased from about 20,000 to 150,000 lb. weight and schedule speed for passenger trains has increased from 30 m.p.h. to as high as 70 m.p.h.

The new conditions arising from longer and heavier trains have led to important improvements in both freight and passenger traffic. The demands for electric traction brakes have likewise led to marked improvements in equipment for that service. For much of the noteworthy developments in the past 10 years, full credit should be given to the pioneer work of engineers and inventors, especially to Mr. Walter V. Turner of the Westinghouse Air Brake Co.

The following are among the important features in the recent advance: (1) The development of entirely independent or simultaneous operation of train and locomotive brakes as may be desired; (2) ability to apply and release the brakes without serious shocks; (3) development of the Clasp Brake, that is, a brake system with two brake shoes on one wheel; and (4) electro-pneumatic control giving simultaneous application of each brake on a train, regardless of the number of cars in the train, the electric being added to the pneumatic features to accomplish this result, and this development applied under the name Universal Control (or UC) Equipment.

In 1869 a locomotive and six cars could be stopped from 60 miles an hour in 2000 ft. by the "straight air" brake. To-day the same train can be stopped in less than one-half that distance.

WHAT STOPS A MOVING TRAIN WITHOUT SKIDDING?

A complete answer to this question is by no means simple. A train at say 50 miles an hour running in still air on a level straight track with throttle valve closed, will stop in a certain distance without the application of brakes, the resisting forces being the air, internal friction of the moving parts and friction of flanges against the rail. The sum of these forces is small compared with

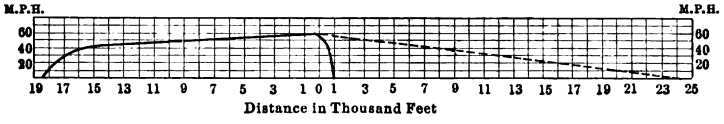


FIG. 36.—Curves of acceleration and deceleration.

Class E2d locomotive. Total weight of train, 559.8 tons. Acceleration distance, 18,500 ft. Time of acceleration, 5 min. 47 sec. Deceleration distance, 954 ft. Time of deceleration, with brakes, 18.7 sec. Broken line represents stop on level track without the use of brakes, obtained by assuming 9.6 lb. per ton retardation due to wind resistance and journal friction. Total energy stored in train and overcome by brake = 126,500,000 foot pounds.

the resistance offered either by the application of brakes or by a considerable up-grade, which facts are well brought out by a study of Fig. 36. Every moving body is capable of doing work before its motion can be diminished or stopped and the amount of work may be measured in foot-pounds or its equivalent as in heat units. A brake shoe pressed against a revolving wheel causes a resistance called dynamic or kinetic friction, the amount of which depends

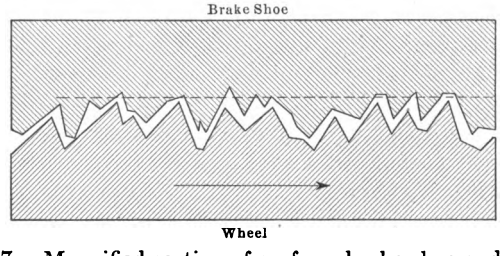


FIG. 37.—Magnified section of surface, brake shoe and wheel.

primarily upon the pressure and upon the nature of the surfaces in contact.

Figure 37 shows how the uneven surfaces of the shoe and wheel may interlock, causing a high coefficient of friction. In order that the wheel shall move, the projections of either or both the wheel and shoe must be sheared off, or the uneven surface of one metal in contact must move over the surface of the other,

causing a high coefficient of friction and a correspondingly rapid rise in temperature. The metal of the shoe in contact with the wheel must not be too hard, otherwise the metal in the tread of the wheel will be rapidly worn away. It is cheaper to wear out the shoe than it is to wear down the wheel. Furthermore, the metal in the shoe is in contact with the wheel continuously during braking and the energy is absorbed by the small area in direct contact with the wheel, resulting in a rapid rise in temperature. This continued, the force required to tear away the metal is reduced and the coefficient of friction correspondingly lowered.

"Unquestionably the constantly changing temperature of the contact surface has an important relation to the force of retardation developed by the tearing down of the metal particles. The resistance due to abrasion is dependent on the ultimate strength of the cast iron. It is well established that above a critical temperature (approaching 900°F.) the ultimate strength of cast iron decreases rapidly and that at or above red heat temperatures (1400° to 1800°F.) its ultimate strength is greatly reduced.

"The force of retardation due to abrasion and the corresponding mean coefficient of friction will therefore be a function of the constantly changing (but always high) temperature of the working metal. Obviously the lower the mean temperature can be maintained, the higher will be the mean coefficient of friction as long as this average temperature is above the critical temperature at which the ultimate strength of cast iron is a maximum.

"Unless the brake shoe is of such a nature or in such a condition that a large proportion of its face area is in working contact with the wheel, and remains so, the principal factor in producing high friction for any given braking condition appears to be the frequent shifting of the bearing area from the heated to the cooler spots over the face of the shoe. If a shoe has a relatively small bearing area, that cannot shift, the wheel is forced to wear rapidly away the highly heated and ineffective shoe metal, and successively exposes cooler metal, which is better able to offer resistance and absorb energy.

"If the friction characteristics developed by a given brake shoe under a given combination of conditions depend, as appears probable from the foregoing considerations, upon the frequent shifting of the contact areas of the shoe surface, the instantaneous as well as the average values of the frictional resistance of the brake shoes during a stop are functions of: (*A*) The fit of the shoe to the wheel; (*B*) the flexibility of the shoe; (*C*) the available bearing area." (S. W. Dudley, Journal A. S. M. E., Nov., 1914.)

Brake Shoes.—In connection with the design of brake rigging, the brake-shoe friction is of primary importance. During the years 1906 to 1914 brake shoes of different material and construction have been tested on the Master Car Builders' Machine at Purdue University, the results of which have been presented before the M. C. B. Association.

Determinations have been made of the average coefficient of friction, the rise in the coefficient of friction at the end of the

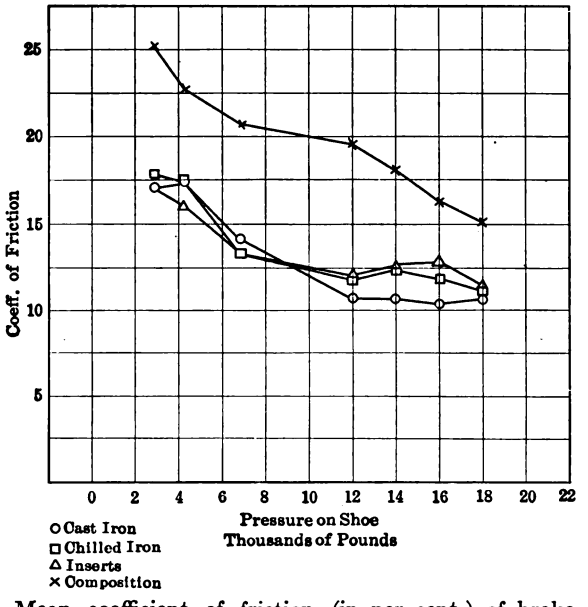


FIG. 38.—Mean coefficient of friction (in per cent.) of brake shoes on steel tired wheels at 65 m.p.h.—Purdue tests.

run, and the loss of weight of the shoe in comparison with the number of foot-pounds of work done. The shoes have been tested at various speeds ranging from 20 to 80 m.p.h., and under pressures of application varying from 1080 lb. to 2000 lb.

The materials have been classified as follows:

(*a*) Plain cast iron such as can be easily drilled.

(*b*) Chilled cast iron where the surface has been partially or wholly hardened by chilling.

(*c*) Cast-iron shoes with inserts of harder material.

(*d*) Shoes having a cast-iron shell filled with a comparatively soft composition.

The readings for each group have been averaged and these averages plotted.

It is apparent from Figs. 38, 38*a* and 38*b* that the coefficient of friction diminishes as the pressure on the shoe is increased, but that for pressure from 12,000 to 18,000 lb., inclusive, the difference is slight. It is furthermore apparent that pressures in excess of 18,000 lb. are not economical.

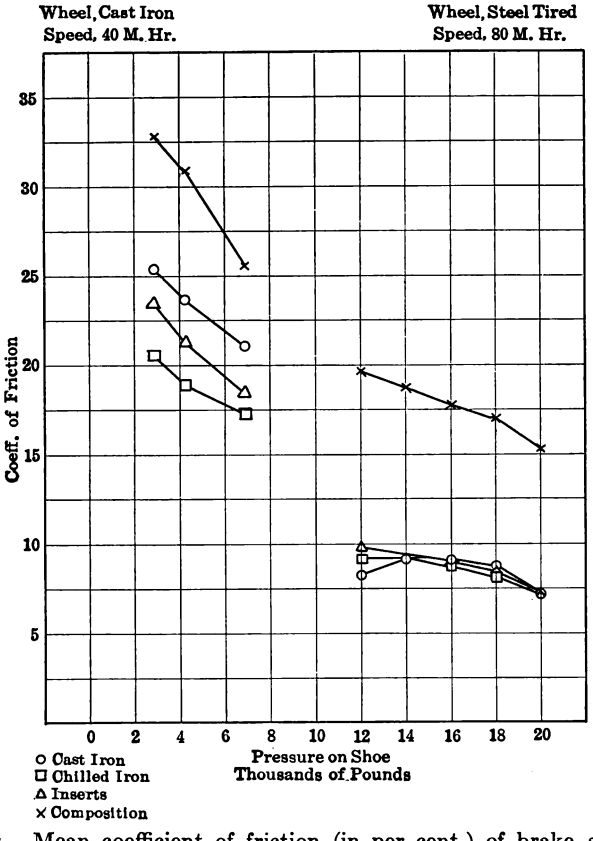


FIG. 38*a*.—Mean coefficient of friction (in per cent.) of brake shoes on cast iron and steel tired wheels at 40 and 80 m.p.h., respectively.

Reference to Fig. 38*a* shows that the coefficient of friction at high speeds is very much less than at moderate speeds, the average coefficient at 80 m.p.h. being less than 10 per cent. or less than one-half the corresponding average at 40 m.p.h.

The coefficient of friction of filled or composition shoes is in

all cases considerably greater than the average for the other three groups, being from 50 to 100 per cent. in excess.

General conclusions cannot safely be drawn as to the effect of speed and pressure on the loss of weight, except that pressures in excess of 18,000 lb. cause an abnormal loss. The general indication is, however, an increasing loss of weight with increase of

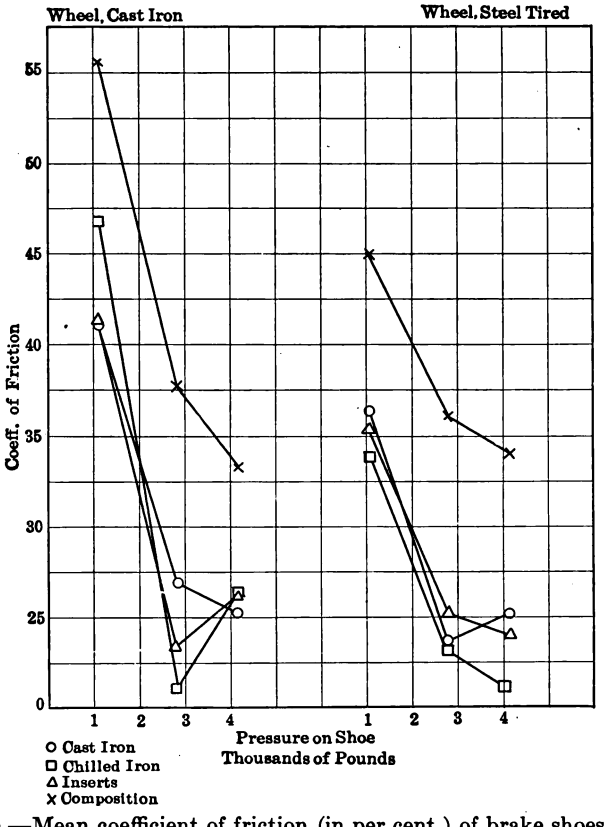


FIG. 38*b*.—Mean coefficient of friction (in per cent.) of brake shoes on cast iron and steel tired wheels at 20 m.p.h.

pressure and speed. In other words, as these two factors increase, the wear of the shoe compared with the work done in stopping the wheel increases.

A comparison of the shoe wear per 100,000,000 ft.-lb. of work done under single shoe and clasp-brake conditions is shown in the following table taken from results of the 1913 Absecon tests:

COMPARISON OF SINGLE- AND CLASP-BRAKE SHOE WEAR PER 100 MILLION FOOT-POUNDS WORK DONE

Brake conditions	Kind of shoes plain		Ratio of wear solid to slotted, per cent.	Ratio of wear single solid to clasp solid, per cent.	Ratio of wear single slotted to clasp slotted, per cent.
	Type "A" solid	Type "B" slotted			
Single	4.382	3.921	111.7	141.1	133.5
Clasp	3.105	2.937	105.9	.....	.....

A comparison of plain solid and plain slotted brake shoes under single- and clasp-brake conditions shows:

A. That the superior durability of the plain slotted shoe as compared with the plain solid amounts to 11.7 per cent. under single-shoe-brake conditions and 5.9 per cent. under clasp-brake conditions.

B. That with plain solid shoes the durability will be increased 41.1 per cent. under clasp-brake conditions as compared with that under single-shoe conditions.

C. That with plain slotted shoes the durability will be increased 33.5 per cent. under clasp-brake conditions as compared with that under single-shoe conditions.

During an application of the brakes, the wheels from instant to instant are at rest at their point of contact with the rail and will remain so until the resistance at the rail is less than the resistance caused by the pressure of the brakes on the wheels. The resistance at the track is also made up of two factors, the weight of the wheel on the rail and the coefficient of static friction. Therefore a wheel becomes locked and it slides because the weight on the wheel times its coefficient of friction between the wheel and rail is less than the pressure on the brake shoe times the coefficient of kinetic friction between the brake shoe and the wheel.

A Problem Worked.—Suppose a steel car weighing 120,000 lb. moving at 50 m.p.h. has two four-wheel trucks, so that each wheel supports 15,000 lb. The coefficient of static friction between the wheel and the rail may be as low as 0.12 or as high as 0.33. Assuming, 0.25, for dry rail and average service, the resistance along the line of the rail to prevent sliding is $15{,}000 \times 0.25 = 3750$, which force must be overcome before the wheel "skids" or slides as shown in Fig. 39.

At 50 m.p.h. the coefficient of friction between a cast-iron brake shoe and a steel-tired wheel at the instant of application has been found to be approximately 0.12. The present nominal braking force on a Pennsylvania R. R. 70-ft. steel passenger car weighing 60 tons, for service application is 80 per cent. of its light weight, based on 60 lb. brake-cylinder pressure and using the quick action brake with high-speed reducing valve. $120{,}000 \times 0.80 = 96{,}000$ lb. which gives a brake-shoe pressure for each of the eight shoes, of $96{,}000 \div 8 = 12{,}000$ lb. $12{,}000 \times 0.12 = 1440$ lb. acting to prevent rotation and 3750 lb. acting to prevent skidding. The force applied by the brake to skid the wheel would be $3750 \div 0.12 =$

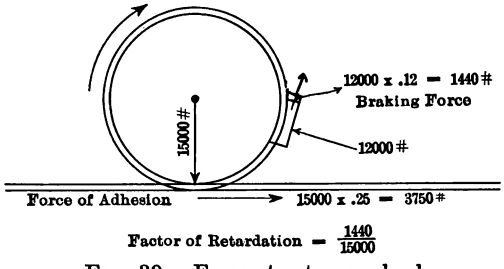


FIG. 39.—Forces to stop a wheel.

31,250 lb. or $31{,}250 \times 8 = 250{,}000$ lb. for the eight wheels. $250{,}000 \div 120{,}000 \times 100 = 208$ per cent. braking force to skid the wheels. This is on the safe side even for emergency application for which 113 per cent. is the nominal braking force with brake-cylinder pressure of 85 lb. This value of 113 is obtained from the proportion,

$$85 \text{ (lb.)} : 60 \text{ (lb.)} = 80 \text{ (per cent.)} : x \text{ (per cent.)} \quad x = 113$$

It will be evident from this calculation that there is a margin from 208 to 113, or 84 per cent. increase possible in the braking force. Since 12,000 lb. pressure per shoe (as here taken) in service application means $12{,}000 \times 80 \div 60 = 17{,}000$ lb. per shoe in emergency, and experience has shown that this is close to the practical limits for the pressure on a single standard shoe for good wearing conditions, it is evident that the only way remaining to obtain more nearly the maximum allowable pressure of the shoe on the wheel before skidding occurs, is by the addition of a second shoe. It has been recently developed in what is

called the "clasp-" brake lever arrangement which is shown in Fig. 45.

The coefficient of kinetic friction changes with the speed. At the instant of applying the brakes, the average coefficient of friction as found in the famous Westinghouse-Galton tests, is given in the following table:

TABLE IV.—COEFFICIENT OF FRICTION AT VARYING SPEEDS. CAST-IRON BRAKE-BLOCKS ON STEEL TIRES

Velocity, miles per hour	Mean coefficient of friction	Velocity, miles per hour	Mean coefficient of friction
60	0.074	25	0.166
55	0.111	20	0.192
50	0.116	15	0.223
45	0.127	10	0.242
40	0.140	7½	0.244
35	0.142	Under 5	0.273
30	0.164	Just moving	0.330

In the problem at hand in order to skid the wheels the coefficient of static friction must equal 3750 ÷ 12,000 = 0.313 which is less then 0.330 given in the table and the wheel would not skid as it nears a stop under the condition assumed.

The above values for the coefficient assume the metals to be cool. As the brakes are applied, the metals in contact, especially the shoe, heats up and the coefficient becomes less as the time of application is lengthened.

Not only is the coefficient changing for each speed but the coefficient uniformly decreases the longer the shoe rubs against the wheel at any given speed, which with the data at hand make it out of the question to fix upon scientifically correct values for the different speeds. The following are recommended as safe values for use in calculations:

Stop, from speed, miles per hour	Mean coef. friction
20	0.15
40	0.11
60	0.09

Forces Acting to Retard a Train.—There is stored in a moving body a certain amount of energy which would be absorbed if the body was suddenly stopped by a solid wall or was otherwise retarded. A train running at 60 m.p.h. will give up the same

amount of energy as if it fell from a height of 120 ft.; at 40 m.p.h. the equivalent drop is 53.5 ft. Thus a six-car train coming to a service stop from 60 m.p.h. in some 1500 ft. dissipates over 132,000,000 ft.-lb. of energy, without harm or disturbance to the passenger.

Let K be the kinetic energy in foot-pounds of a body weighing W lb. In falling a distance h ft. the work done will equal Wh ft.-lb. In falling, the body will acquire a velocity v ft. per sec. in a time t sec.; $v = gt$ where $g = 32.16$ or $t = \frac{v}{g}$; but h is the distance the body falls and equals $\frac{1}{2}\ vt$.

Hence $h = \frac{1}{2}\ gt^2$ and $K = \frac{1}{2}\ gt^2 \times W$ or $K = \frac{Wv^2}{2g} = FS =$ the work done where F is the force acting to produce the velocity in S ft.

$$F = \frac{Wv^2}{2gS} \text{ or } S = \frac{Wv^2}{2gF}$$

If F is the retarding force, the above equation also applies.

Let V = velocity m.p.h., $v = \frac{5280}{60 \times 60} V = 1.446V$ or $v^2 = 2.15V^2$. To express braking force F in terms of per cent. of W lb. braked, find the value of F for 100 lb. of weight W or substitute 100 for W, and call this percentage of force, "F per cent."

$$F \text{ per cent.} = \frac{2.15V^2 \times 100}{2 \times 32.16 \times S} = 3.34\ \frac{V^2}{S} \qquad (51)$$

This does not include the extra retarding force necessary to offset the effect of the rotating wheels, which has been found to approximately equal the wind resistance at different velocities so that the one is considered to neutralize the effect of the other.

Let V_2 be the higher and V_1 the lower velocity. Then Eq. 51 becomes

$$F \text{ per cent.} = 3.34 \times \frac{V_2^2 - V_1^2}{S} \text{ or } S = 3.34\ \frac{V_2^2 - V_1^2}{F \text{ per cent.}} \qquad (51a)$$

If F is the factor of retardation, expressed in decimal equivalent rather than in per cent., the equation reads, $F = 0.0334\ \frac{V_2^2 - V_1^2}{S}$. Attention may be called again to the fact that the equations for uniform acceleration hold equally true for uniform deceleration.

Example.—Find the calculated distance to stop a passenger car of light weight, 100,000 lb., brake to 90 per cent. of its light

weight from 60 m.p.h., assuming coefficient of friction as 0.10

$$100{,}000 \times 0.90 \times 0.10 = 9000 \text{ lb. retarding force.}$$

$$\frac{9000}{100000} = 0.09 = \text{factor of retardation} = 0.09 \times 100 = 9.0 = \text{factor in per cent.} = F \text{ per cent.}$$

$$S = 3.34 \times \frac{60 \times 60}{9} = 1336 \text{ ft.}$$

The Absecon tests in 1913 gave an emergency stop with clasp brakes from 60 m.p.h. in 725 ft.

By the same method of procedure to find the time in seconds

$$t = 4.78 \frac{V_2 - V_1}{F \text{ per cent.}} \tag{52}$$

The factor of retardation being different for cars and locomotive it is desired to find the length of stop of a train taking into account these conditions. Returning to Eq. 51*a* and using the subscript *c* for car and *l* for locomotive (including tender), then for *N* cars

$$NF_c = \frac{0.0334 \times V^2}{S_c} \cdot W_c$$

Where F_c is the factor of retardation for the cars.
For the locomotive,

$$F_l = \frac{0.0334 \times V^2}{S_l} W_l$$

Knowing W_l, W_c, S_l and S_c, the distance S to stop the entire train weighing W lb. becomes,

$$S = \frac{WS_lW_c}{S_cW_l + NS_lW_c} \tag{53}$$

Eq. 51*a* may be used for a stop on a grade, as follows:

$$\text{For up-grade, } S = 3.34 \frac{V_2^2 - V_1^2}{F \text{ per cent.} + G} \tag{54}$$

$$\text{And for down-grade, } S = 3.34 \frac{V_2^2 - V_1^2}{F \text{ per cent.} - G} \tag{54a}$$

Where G is the grade in per cent., Eq. 51*a*, 54 and 55, theoretically correct for uniform retardation during the change of speed from V_2 to V_1, can at the best be only approximate on account of the variation in the coefficient of friction for different speeds. Moreover, deceleration curves (see Fig. 41) plotted on speed-dis-

tance axes is not a straight line throughout, but is more nearly so down to 30 m.p.h., Eq. 51 or 51*a* may be applied twice, (1) to the part of the curve above 20 or 30 m.p.h. which is nearly straight and (2) below 20 or 30 m.p.h. to a stop, using in each case the average value of the coefficient of friction between shoe and wheel for each distance. This will give the best stop under the conditions assumed.

Nominal Braking Ratio.—The foregoing does not include the question of leverage ratio or nominal braking ratio, which must be included to complete the analysis of the problem.

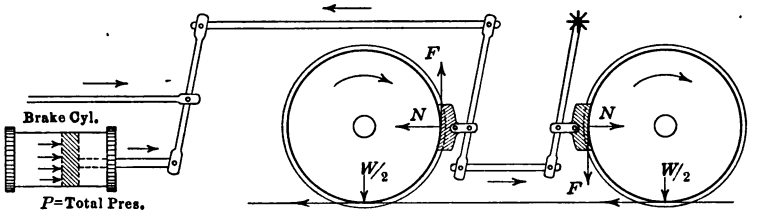


FIG. 40.—Diagram illustrating braking ratio.

Let L = leverage ratio.
E = efficiency of brake rigging.
P = total pressure on piston of brake cylinder.
W = wt. in lb. on wheel or wheels considered.
f = mean coefficient of brake-shoe friction.
S = distance in feet during deceleration, as in Eq. 51*a*.
V_2 = higher speed, m.p.h.
V_1 = lower speed, m.p.h.
B = nominal braking ratio.
$PL = BW$.

B is assumed from experience at certain values depending on the class and weight of vehicle, speed, grade conditions, type of brake, type of wheel and shoes, etc. Common values are: Freight service, 60 per cent.; passenger cars, 80 or 90 per cent. locomotives, 60 per cent. on drivers, 45 per cent. on trailers and trucks, 80 per cent. on tender. Other average values are given later under Brake Design; its true value under any given set of conditions may be determined as follows, expressing B, E and f in decimals:

$$N = BW = PLE. \quad \text{(See Fig. 40.)}$$

$$\text{Braking force} = F = Nf = PLEf = BWEf = 0.0334\left(\frac{V_2^2 - V_1^2}{S}\right)W.$$

$$\therefore B = 0.0334\,\frac{V_2^2 - V_1^2}{SEf} \text{ (on level track)} \qquad (55)$$

If the conditions involve the stopping in distance S on a down grade of p per cent.:

$$F = BWEf = 0.0334\left(\frac{V_2^2 - V_1^2}{S}\right)W + p(p \text{ in decimals})$$

$$\therefore B = \frac{1}{Ef}\left[\frac{0.0334(V_2^2 - V_1^2)}{S} + p\right]\text{(down grade)} \qquad (55a)$$

For up grade, substitute $-p$ for $+p$ in (55a). The values of S and Ef must be chosen according to the most exacting or dangerous conditions to be encountered. S depends on the speed, frequency of stops, minimum distance between signals, etc. For passenger service, usually, emergency stops should be made from 60 m.p.h. in from 1000 to 1200 ft. The most probable values of Ef are given in Table V, page 99.

If K represents the time in seconds from the point of application of brakes to the equivalent point of instantaneous application, then

$$B = \frac{1}{Ef}\left[\frac{0.0334(V_2^2 - V_1^2)}{S - 1.467K_1V_2} + p\right]\text{(down grade)} \qquad (55b)$$

since V_2 miles per hour equals $1.467V_2$ ft. per sec. It is not always practicable to use this equation as the quantity K_1 varies considerably, especially with the different types of valve mechanisms forming part of the brake equipment. It may usually be taken from 1.5 sec. for 45 m.p.h. to 2 sec. for 60 m.p.h. in passenger service.

To find the length of stop S, with a certain braking ratio B, these equations may be solved for S as follows:

$$S = 0.0334\,\frac{V_2^2 - V_1^2}{BEf} \text{ (level track)} \qquad (56)$$

$$= 0.0334\,\frac{V_2^2 - V_1^2}{BEf - p} \text{ (down grade)} \qquad (56a)$$

$$= 1.467K_1V_2 + \frac{0.0334(V_2^2 - V_1^2)}{BEf - p} \text{ (down grade)} \qquad (56b)$$

The following values of $E \times f$ were taken from results in the Pennsylvania-Westinghouse tests of 1913.

TABLE V.—VALUES OF $E \times f$

Kind of brake rigging		Clasp brake		Single shoe[1]	
Type of brake shoe		Plain	Flanged	Plain	Flanged
Speed, m.p.h.	Braking ratio				
30.....	125	0.141	0.169	0.108	0.112
	150	0.129	0.154	0.099	0.103
	180	0.118	0.141	0.090	0.094
60.....	125	0.103	0.122	0.074	0.090
	150	0.094	0.112	0.068	0.082
	180	0.086	0.102	0.062	0.075
80.....	125	0.092	0.109	0.070	0.071
	150	0.084	0.100	0.064	0.068
	180	0.077	0.092	0.059	0.062

[1] Value of data uncertain due to non-uniform brake-shoe conditions.

Equation 55 may be written:

$$f = \frac{0.0334(V_2^2 - V_1^2)}{BES} \tag{57}$$

Example.—Let speed be 60 m.p.h. and total stopping distance 1000 ft. (S_2) for nominal braking ratio of 125 per cent. and 80 per cent. car rigging efficiency, distance from start of braking to equivalent instantaneous application (S_1) be 70 ft., then $S = S_2 - S_1 = 1000 - 70 = 930$ ft., $E = 80$, $B = 1.25$, $V_2 = 60$, $V_1 = 0$.

The mean coef. fric.,

$$f = \frac{0.0334(60)^2}{1.25 \times 0.85 \times 930} = 0.122$$

From the statement regarding the wide variation in value of the coefficient of brake-shoe friction from 60 m.p.h. down to very slow speeds and the confirmation of this from a study of resistance curves as shown in Fig. 41, it is evident that the value of 0.122 is an approximation of an average value and is not, therefore, the value at the beginning of application of the brakes. Moreover, as previously pointed out, the coefficient of friction is modified by the time of application of the brakes, which would reduce somewhat the average value of f, as above determined.

Breakaway Tests.—Fig. 41 is reproduced from a series of tests made in November, 1912, on the Lake Shore and Michigan Southern Railroad. A study of this plot will bring out many points referred to in the previous pages.

The lower curves indicate that the deceleration increases to the point P and then remains nearly constant until near the end of the stop, when it increases very rapidly. The gradual rise at the beginning is due to the time required for the valve mechanism

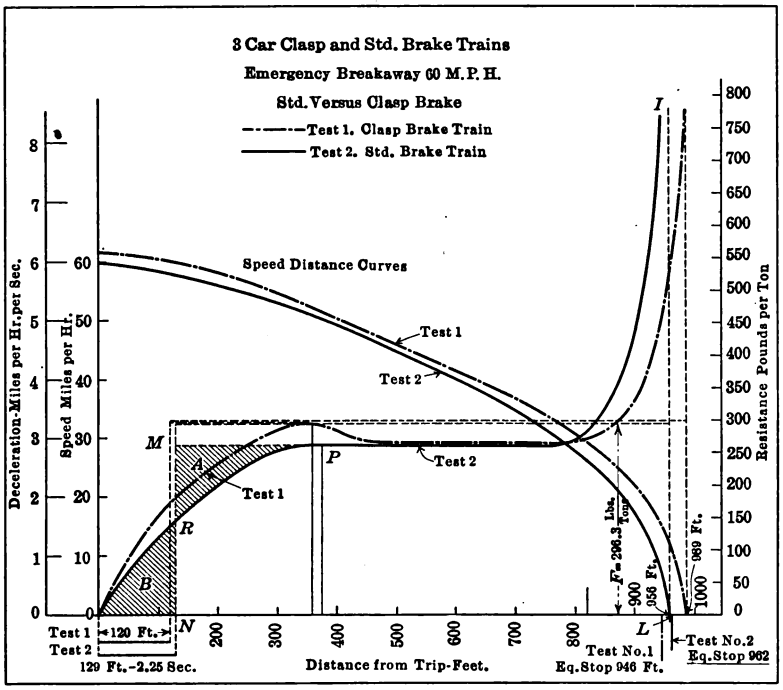


FIG. 41.—A study of brake tests.

This plot shows increased resistance to braking from the clasp brake, during the first 400 ft. of braking. It also illustrates the method of finding the equivalent point of instantaneous application.

to act and for the brake cylinder to build up. Although this occupies only a very few seconds, it is an appreciable element in accurate determination of the length of stop. Since it is impossible to know accurately what the average actual brake cylinder pressure will be (due to this gradual building up at the start), it is more convenient to assume that the maximum pressure is obtained instantly and is held uniformly from the *equivalent*

point of instantaneous application, which is found by prolonging the deceleration curve from P horizontally to the left to a point M such that the vertical line MN will cut the deceleration curve at R, so as to make the area A equal to the area B. The point N is the equivalent point of instantaneous application, for it is evident that the area $LNMPIL$ is equal to the area $LOPIL$ and such areas represent the amount of work done. Therefore, if the train could run at the initial speed V_2 to N and then the brakes

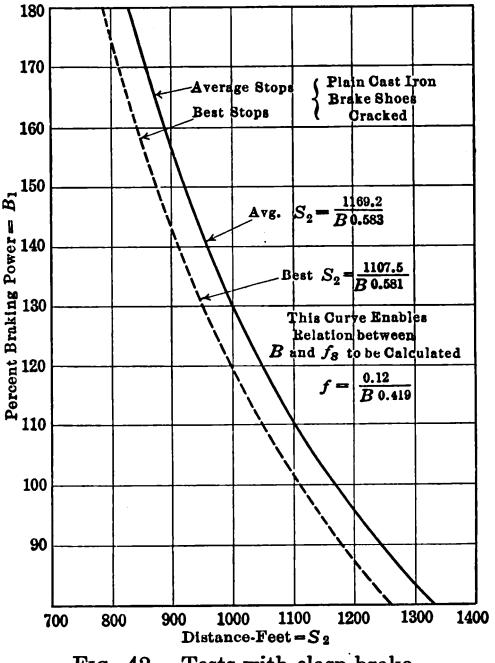


FIG. 42.—Tests with clasp brake.

Shows lengths of stop in single car emergency breakaway stops made with No. 3 clasp brake, from 60 m.p.h. with the electro-pneumatic equipment.

instantly applied with maximum force, it would stop at L just the same as in the actual stop. The distance ON varies with different types of brake mechanism and also with different speeds.

The Pennsylvania-Westinghouse Tests of 1913 gave single car emergency breakaway stops, as shown in Fig. 42 with No. 3 clasp brake. The length of stop S_2 in terms of per cent. braking power B, as illustrated, may be expressed by an equation of the general form: $S_2 = \frac{K}{B^x}$, where K is a constant depending on S_2

and B. The relation of S_2 and B are shown for the "best" and for "average" stops. A fair average for f_1 (the mean coefficient of friction) from 60 m.p.h. is 0.10.

Problem 11.—Find the length of stop, according to equation for average stop, in Fig. 42, using 120 per cent. braking power, and compare results with Fig. 42.

BRAKE LEVERS

Two of the more common systems of brake levers are shown in Figs. 43 and 44. Suppose that the pressure on each of the four brake beams has been calculated to be 14,000 lb., so that with a leverage ratio of 8.5, the nearest standard size cylinder will be 12 in. by 12 in. at 60 lb. pressure. Calculations for stresses in

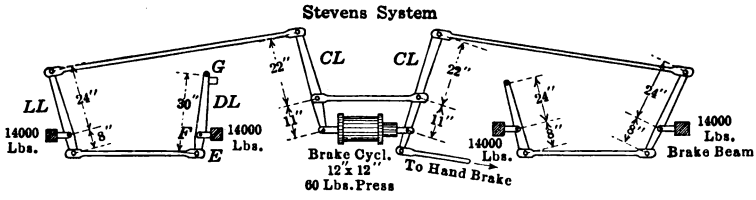


FIG. 43.—The Stevens system of levers.

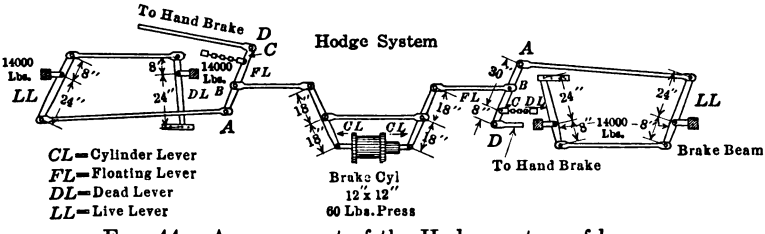


FIG. 44.—Arrangement of the Hodge system of levers.

rods and levers are based on the simple law of leverage, that a force times its lever arm equals the opposing force times its lever arm.

Problem 12.—In the Stevens brake system, using data in Fig. 43, find (*a*) the point F of attachment of brake beam, if total length of the dead lever FG = 30 in.; (*b*) find stresses in each rod.

Problem 13.—In the Hodge system, Fig. 44, (*a*) find the point of attachment B on the floating lever DA, if AD = 30 in.; (*b*) stresses in rods; (*c*) distance CD, if pull on hand brake rod is limited to 1200 lb.

In Fig. 46, page 111, is shown the outline diagram of lever arrangements of the four brake riggings used on the noteworthy Pennsylvania-Westinghouse brake tests of 1913.

BRAKE DESIGN

Problem 14.—Calculate the diameter of the brake cylinder to stop a car from 60 m.p.h. in 1000 feet by emergency application.

Notation

E = efficiency of brake rigging.
f = coefficient of friction between brake shoe and wheel.
W = weight in pounds of the vehicle considered.
W_l = weight in pounds of locomotive.
W_t = weight in pounds of train.
W_c = weight in pounds of cars.
F = retarding force in pounds.
P = total brake-shoe pressure per car.
p = pressure per square inch in brake cylinder.
L = leverage ratio.
D = diameter of brake cylinder, inches.
A = area of brake cylinder, square inches.
a = deceleration, or average rate of change of velocity during braking.
r = factor of retardation $= \dfrac{\text{total breaking force} = F}{\text{weight on wheels braked} = W}$.
r_c = factor of retardation for cars.
r_l = factor of retardation for locomotive.
V = velocity in feet per second.
t = time in seconds.
S = distance in feet.
B = nominal braking ratio. Should be decimal equivalent.

Assumptions

$W_t = 3W_l$ or $W_c = 2W_l$ or $W_t = \frac{3}{2}W_c$.
$r_l = 0.50r_c$
E = 85 per cent.
Ratio L = 9 to 1.
f = 10 per cent.
V_o = 60 m.p.h. (88 ft. per sec.)
a = 0.256 ft. per sec. per sec.

Train of one locomotive and six cars.

Train to be stopped in 1000 ft. with emergency application p = 105 lb. Lading: sleepers 3 per cent.; coaches, 10 per cent.; baggage and other cars on passenger trains, 10 per cent.

Calculations

(*a*) To find the space passed over during the time in making full application of the brakes from 60 m.p.h. $S = Vot - \frac{1}{2}at^2$; a = 0.256 ft. per sec.

per sec.; S = space passed over (in 2 sec.) in which maximum effort is being applied; $S = 88 \times 2 - \frac{1}{2} \times 0.256 \times 4 = 176$ ft. (nearly).

(*b*) To find the factor of train retardation in terms of car retardation. The general principle for this calculation is that the product of the weight of the train times its factor of retardation equals the sum of the weight of the locomotive times its factor of retardation plus the weight of the cars times its factor of retardation, that is:

$$r_t = \frac{r_l \times W_l + r_c \times W_c}{W_t} \tag{58}$$

$$r_t = \frac{0.50r_c \times \frac{1}{2}W_c + r_cW_c}{\frac{3}{2}W}.$$

or

$$r_t = \frac{5}{6}r_c = 0.833r_c \tag{59}$$

(*c*) To find the retarding force in per cent. of the car weight

$$\text{Work} = F \times S = \frac{Wv^2}{2g}$$

or

$$\frac{F}{W} = \frac{V^2}{2gS} = N \tag{60}$$

Assume a coach with 10 per cent. leading, then the factor of retardation for the train,

$$r_t = \frac{F}{W} = \frac{1.10 \times 88 \times 88}{64.4 \times (1000 - 176)} = 16.1 \text{ per cent.}$$

But,

$$r_t = 0.833r_c$$

hence,

$$r_c = 19.3 \text{ per cent.}$$

This is the equivalent uniform percentage of retarding force in terms of the light weight of the car under the conditions referred to above

(*d*) To find the diameter of the brake cylinder with an average coefficient of brake shoe friction of 10 per cent., it would require $\frac{19.3}{0.10} = 193$ per cent. of the light weight of the car.

If P = total brake-shoe pressure per car, then $P = 1.93W_c$.

Assume $P = 105$ lb.; $L = 9$ to 1; $E = 0.85$:

$$P = W_c \times r_c = p \times A \times L \times E = p \times \tfrac{1}{4}\pi D^2 \times L \times E \tag{61}$$

from which,

$$D^2 = \frac{1.93W_c}{0.7854 \times p \times L \times E} = \frac{1.93W_c}{0.7854 \times 105 \times 9 \times 0.85}$$

$$D = \sqrt{0.00325W_c} \tag{62}$$

The assumptions and constants in the above are from the report on Train Brake and Signal Equipment before the Am. Ry. M. M. Asso., in 1910.

Brake cylinders are made in standard sizes, 6 in., 8 in., 10 in., etc., in diameter, and the sizes have been fixed for cars of weights between certain limits.

Equation 61 may be solved for any one unknown value if the other values are known or assumed.

Problem 15.—With assumptions as given for the problem just solved, except that train be stopped in 1200 ft., deduce new constant for Eq. 62.

SOME PRINCIPLES OF BRAKE DESIGN

1. Determine the braking ratio by Eq. 55 or 55*a*. The latest practice is to use a 50-lb. cylinder basis for all equipments, except passenger cars. On the latter, 60 lb. is used for all types of triple valves and the universal valve, while 86 lb. is used with the PC equipment. The recommended braking ratio corresponding to these cylinder pressures is as follows, for ordinary conditions:

1. Driver brakes with plain triple valve, 75 per cent. on 50 lb.
2. Driver brakes with distributing valve, 60 per cent. on 50 lb.
3. Trailer and truck brakes with plain triple valve, 60 per cent. on 50 lb.
4. Trailer and truck brakes with distributing valve, 45 per cent. on 50 lb.
5. Tenders with quick-action triple valve, 85 per cent. on 50 lb.
6. Tenders with distributing valve, 80 per cent. on 50 lb.
7. Passenger cars, quick-action triple valves, 90 per cent. on 60 lb.
8. Passenger cars, PC equipment, 90 per cent. on 86 lb.
9. Passenger cars, UC equipment, 90 per cent. on 60 lb.
10. Freight cars, quick-action triple valves, 60 per cent. on 50 lb.

The percentages given above refer to the complete light weight of the vehicle, with the exception of driver brakes, which refer to the weight on drivers, engine in working order. This also applies to the truck and trailer brakes of engines. If, however, the results obtained from Eq. 55 or 55*a* show that the higher braking ratio should be used, this higher ratio should be followed up to the point where an emergency application will be in danger

of sliding wheels. Beyond that it is not safe to go and any higher braking force cannot be obtained.

It is considered proper to figure the braking power for service and let the emergency come where it will, except that the foundation brake gear will have to be designed for the stresses brought about during the emergency application.

2. Calculate the diameter of the brake cylinder which will give the required brake-shoe pressure, using a standard size of cylinder. Experience has shown that a 9 to 1 total leverage is about the highest that is practicable in steam road service, and then only in cases of the lighter cars with the best class of foundation brake gear. This ratio is recommended as a maximum for cars with the center of the shoe not more than 2 in. below the center of the wheel. Where the center of the shoe is from 2 in. to 5 in. below the center of the wheel, it is better to keep the total leverage down to 8 to 1 or less. With very heavy cars and the shoes lower than 5 in. below the center of the wheel, the maximum total leverage of 7 to 1 is considered good practice. This last, however, has been usually applied to cases where the Pullman type of foundation brake gear on six-wheel trucks is used. If the clasp brake is used the centers of the shoes are usually pretty high and their application is such that there is little or no danger of compressing the springs on the trucks, and the maximum leverage of 9 to 1 can be used with such an equipment. The trouble with high leverage is that with low shoes there is so much tendency to compress the truck springs and draw the body of the car down solidly on the truck, which results not only in a very hard riding car, but also in materially lengthening the piston travel of the brake cylinder, which reduces the pressure in the latter, since its volume is greater than figured upon for the auxiliary reservoir size used.

3. Determine the proper auxiliary reservoir volume.

Starting with a brake cylinder of the size required by the vehicle to be equipped, and by a proportioning of the leverage which shall accord with the service required, let

C = volume of brake cylinder, in cubic inches.
P = service equalization pressure, in absolute units.
R = volume of auxiliary reservoir, in cubic inches.
a = absolute initial pressure in the auxiliary reservoir.
r = permissible range of brake pipe reductions.

Then will,

$$r = a - P$$

and by the principle of equalization of pressures,

$$P = \frac{a \times R}{R + C}$$

or

$$R = \frac{P}{a - P} \times C = \frac{P}{r} \times c$$

$\frac{P}{r}$ = ratio of the service equalization pressure.

Assume values from current practice, if $P = 50$, $a = 70$; then $r = 20$.

$$R = \frac{65}{20} \times C = 3\tfrac{1}{4}C$$

Nearly all modern brake equipments have a standard size auxiliary reservoirs recommended by the manufacturers, to equalize with brake cylinder at 50 lb. from initial 70 lb. reservoir pressure. As a rule, these standards may be assumed as correct without further calculation.

(4) Lay out the levers on the Stevens, on the Hodge system, or on some approved modification of these. Calculate the strength of the brake beams for the forces to be applied, using the following M.C.B. standards: Maximum unit stress in levers, 23,000 lb. Maximum unit stress in rods, 15,000 lb., no rods to be less than $\frac{7}{8}$ in. in diameter. Maximum unit stress in jaws, 10,000 lb., pin bearing not to exceed 23,000 lb. per square inch.

The chart, Fig. 45, may be used to facilitate calculations, as shown by the following examples:

Example 1.—A passenger car weighing 80,000 lb. is to be braked 90 per cent. of its light weight, 12-in. cylinders being used, leverage being based on 60 lb. in brake cylinder. What ratio of leverage is required?

First find the braking power in pounds by following the vertical line from 80,000 on the axis marked weight of car until it meets the 90 per cent. braking power diagonal of A, and follow across from this point, as shown by dotted line, to the axis marked Braking Power in Pounds, of B, which represents 72,000 lb. braking power. Now start again with 60 lb. on the axis marked Unit Cylinder Pressures, and follow the horizontal line from that point until it intersects the cylinder diameter diagonal marked

12 in. of C. From C follow a vertical line as shown by dotted line to the axis marked Total Cylinder Pressure, this giving the point D, which represents about 6800 lb. total pressure on the piston. A continuation of the horizontal line through the point B, found above, meets a continuation of the vertical line through the point D of F. E lies between the 10 to 1 and 11 to 1 leverage ratio diagonals, and represents a leverage of about 10½ to 1.

Example 2.—On a 90,000-lb. car 14-in. cylinders are used, and leverage ratio is 8 to 1, based on 60 lb. cylinder pressure. At what per cent. of its weight is this car braked?

Starting with the 60-lb. horizontal line, its intersection with the 14-in. cylinder line is H. A vertical from H intersects the 8 to 1 leverage ratio line of F, as shown by the dotted line. Then follow a horizontal line from F, as shown, until it intersects a vertical line from the point representing the weight of the car (90,000), giving the point G lying between the 80 per cent. and 85 per cent. braking power diagonals; G then represents 82 per cent. braking power.

Problem 16.—What size brake cylinders must be used on a 100,000-lb. car, if it is to be braked at 90 per cent. of its weight, leverage ratio being 8 to 1, based on 60 lb. unit cylinder pressure.

Example 3.—Given a 40,000-lb. car, capacity 100,000 lb. and braking power 60 per cent. of its light weight. What will be the per cent. braking power when loaded with 100,000 lb.?

Follow 40,000-lb. vertical line to 60 per cent. diagonal, giving the point W. When loaded, total weight on wheels is 140,000 lb. From W follow a horizontal line, therefore, until it intersects the 140,000-lb. vertical line at Z, giving 17 per cent. braking power.

THE 1913 PENNSYLVANIA-WESTINGHOUSE TESTS

A comprehensive investigation of air brakes and brake rigging, including tests of clasp brakes and electro-pneumatic control for a passenger train of ten steel cars, was carried out near Absecon, N. J., in 1913 by the Pennsylvania Railroad in cooperation with the Westinghouse Air Brake Co. This scientific study is by far the most important since the days of the noteworthy Galton-Westinghouse trials of 1878. The results of the 1913 tests have been reported by the Test Department (Mr. C. D. Young, Engineer of Tests) Pennsylvania Railroad in a 400-page bulletin, and a carefully prepared digest by Mr. S. W. Dudley, Asst.

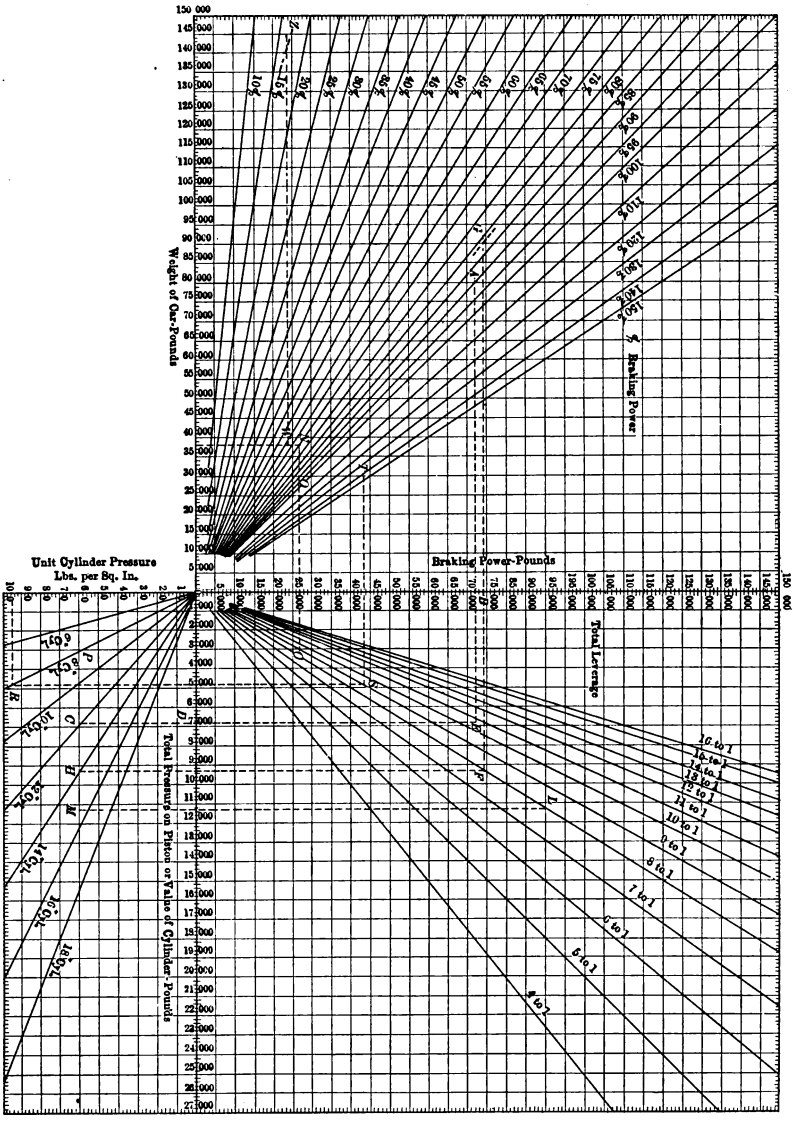


FIG. 45.—Brake leverage chart.

Chief Engineer, Westinghouse Air Brake Co., was printed in the November, 1914, Journal of the American Society of Mechanical Engineers.

Some of the results of these important tests have been referred to earlier in the chapter. Special consideration may be given to the following conclusion:

When the UC equipment is used on the cars an arrangement giving a high emergency braking power on the locomotive, with a blow-down feature, has advantages as follows:

(*a*) Shocks between locomotive and cars practically eliminated.

(*b*) Shorter stops.

(*c*) No more wheel sliding than to be expected with the present installation of ET equipment.

Brake Rigging.—An efficient design of brake rigging must be produced before the advantages of improved air brakes or brake shoes can be fully utilized.

The use of the clasp type of brake rigging (Fig. 45) eliminates unbalanced braking forces on the wheels and so avoids the undesirable troublesome journal and truck reactions that come from the use of heavy braking pressures on but one side of the wheel. This has an important effect not only on freedom from journal troubles but also in enabling the wheel to follow freely vertical inequalities of the track.

The clasp brake also improves the brake-shoe condition materially, both as to wear and variability of performance.

Although the clasp brake rigging will produce better stops than a single shoe brake rigging equally well designed (other conditions being equal), its advantage in this direction is of less importance than in the improved truck, journal and shoe conditions mentioned above.

The tests indicated that at least 85 per cent. transmission efficiency could be obtained with either single shoe or clasp brake rigging.

The following features were observed to be of importance if maximum overall brake rigging efficiency is to be secured:

(*a*) Protection against accidents that may result from parts of rigging dropping on the track.

(*b*) Maximum efficiency of brake rigging at all times to insure the desired stopping with a minimum per cent. of braking power.

(*c*) Uniform distribution of brake force, in relation to weight braked, on all wheels.

(*d*) With a given nominal per cent. braking power, the actual braking power to remain constant throughout the life of the brake shoes and wheels.

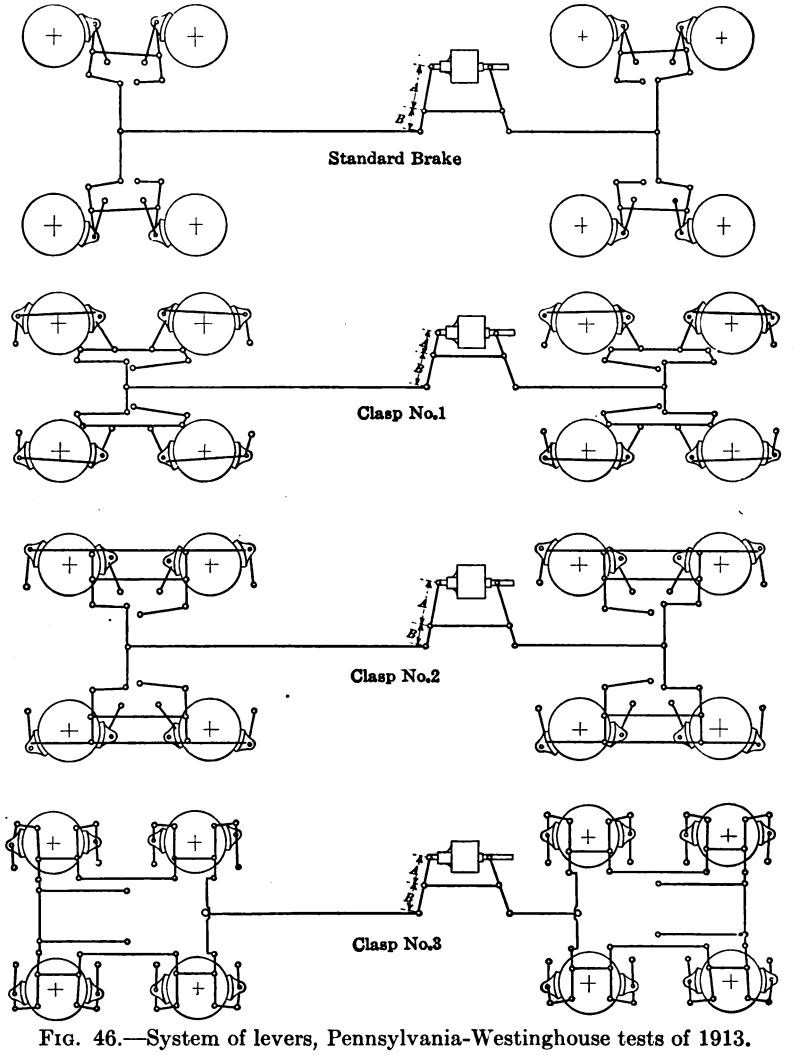


FIG. 46.—System of levers, Pennsylvania-Westinghouse tests of 1913.

(*e*) Piston travel to be as near constant as practicable under all conditions of cylinder pressure.

(*f*) Minimum expense of maintenance and running repairs of brake rigging between the shopping of cars.

STRESSES BETWEEN CARS

We may now proceed to determine the stresses between different cars in a train when the brakes are applied. Such a calculation has been undertaken by Mr. C. H. Beck of the Westinghouse Traction Brake Co. Assume, first, a train to consist of two cars, a loaded car ahead and an empty car in the rear; let

W = total weight of train.
W_1 = total weight of loaded car.
W_2 = total weight of empty car.
F = retarding force of train.
F_1 = retarding force of loaded car produced by brake-shoe pressure.
F_2 = retarding force of empty car produced by brake-shoe pressure.
f = retardation of train.
f_1 = retardation of loaded car separately.
f_2 = retardation of empty car separately.

Then from the fundamental equation, force = mass × acceleration (or deceleration).

$$f = g\frac{F}{W} \tag{63}$$

$$f_1 = g\frac{F_1}{W_1} \tag{64}$$

$$f_2 = g\frac{F_2}{W_2} \tag{65}$$

The difference between (f) and either (f_1) or (f_2) represents the change in retardation caused in that car by being coupled to the other in making a stop. It will be noted that the value of (f) will lie between the values of (f_1) and (f_2).
Let

$$f - f_1 = r_1 \tag{66}$$

and

$$f_2 - f = r_2 \tag{67}$$

(r_1) and (r_2) represent the change in retardation produced on the loaded and light cars, respectively.

If accelerated or retarded motion of the train is caused by force R

$$R = \frac{W}{g}f \tag{68}$$

The change in retardation in the loaded car is represented by r_1 and, similarly, the change in retardation in the empty car is r_2. This change in retardation may be substituted in Eq. 68, whence,

$$\begin{aligned} R_1 &= \frac{W_1}{g} r_1 \\ &= \frac{W_1}{g} (f - f_1) \\ &= \frac{W_1}{g} \left(g \frac{F}{W} - \frac{F_1}{W_1} \right) \\ &= W_1 \left(\frac{F}{W} - \frac{F_1}{W_1} \right) \end{aligned} \tag{69}$$

similarly

$$\begin{aligned} R_2 &= \frac{W_2}{g} r_2 \\ &= W_2 \left(\frac{F_2}{W_2} - \frac{F}{W} \right) \end{aligned} \tag{70}$$

In any given case R_1 and R_2 are equal and it is necessary to solve for one or the other only, to obtain the value of drawbar stress between the two cars mentioned or between two sections of a train.

The value of F_1 and F_2, representing the retarding force, is produced by the brake-shoe pressure (P) times the coefficient of friction μ, or:

$$F_1 = P_1 \mu_1 \tag{71}$$

and

$$F_2 = P_2 \mu_2 \tag{72}$$

The brake-shoe pressure may readily be made constant throughout any given stop, but the coefficient of friction is found from experiment to be subjected to quite a wide range as influenced by numerous conditions, principally that of change in speed.

However, the same change in speed occurs on all cars in a train at approximately the same time, if the train remains intact, and if we assume a definite value for the coefficient of friction between the limits of its range, it is reasonable to suppose that this value is realized at some point during the stop, and the corresponding drawbar strain is thereby produced.

Expanding the two-car train taken as an example to an assumed 100-car train with a locomotive and 50 loaded cars ahead and 50 empty cars in the rear, and assuming a coefficient of friction of

0.10, through use of Eq. 69, a table is produced, giving the drawbar pull throughout the train, during a stop, at the point when the coefficient of friction named above was realized.

Data:	Weight of locomotive	200,000 lb.
	Weight on drivers	187,000 lb.
	Light weight of tender	100,000 lb.
	Weight of tender with load	170,000 lb.
	Total weight of locomotive and tender	370,000 lb.
	Weight of empty car	40,000 lb.
	Weight of loaded car	140,000 lb.

Taking conservative values:
Braking power of locomotive.....60 per cent. of weight on drivers.
Braking power of tender.....90 per cent. of light weight.
Braking power of cars (freight)..... 60 per cent. of light weight.
Retarding force of locomotive..... $=187{,}000\times0.60\times0.10=11{,}220$ lb.
Retarding force of tender..... $=100{,}000\times0.90\times0.10=9{,}000$ lb.
Total retarding force of locomotive and tender..... $=20{,}220$ lb.
Retarding force of cars..... $=40{,}000\times0.60\times0.10=2{,}400$ lb.
Total weight of train (W) $=370{,}000+50\times140{,}000+50\times40{,}000=9{,}370{,}000$ lb.
Total retarding force of train (F).....$20{,}220+100\times2400=260{,}220$ lb.

With reference to the point between the tender and first car, from Eq. 69:

$$R_1 = W_1 \frac{F}{W} - F_1$$

Substitution in the above equation:

		Lb.
W_1 = weight of locomotive and tender	=	370,000
F = retarding force of whole train	=	260,220
W = weight of whole train	=	9,370,000
F_1 = retarding force of locomotive and tender	=	20,220

Therefore

$$R_1 = 370{,}000 \left(\frac{260{,}220}{9{,}370{,}000}\right) - 20{,}220 = 10{,}275 - 20{,}220$$

$$= -9945 \text{ lb.}$$

the algebraic sign (−) indicating compression.

Similarly, with reference to the point between the first and second cars:

		Lb.
W_1 = weight of front section = 370,000+140,000	=	510,000
F = retarding force of whole train	=	260,220
W = weight of whole train	=	9,370,000
F_1 = retarding force of front section = 20,220 × 2,400	=	22,620

Therefore,

$$R_1 = 510{,}000 \times \frac{F}{W} - 22{,}620 = 14{,}162 - 22{,}620 = -8458 \text{ lb.}$$

The curve representing the normal drawbar stresses produced as above is shown in Fig. 47, from which the characteristics may be more readily noted. A compressive force is produced between cars up to the eighth car, where the force changes to tension and continues so throughout. The maximum force occurs between the fiftieth and fifty-first cars amounting to 64,444 lb., a greater force than the drawbar pull of the locomotive. The compression

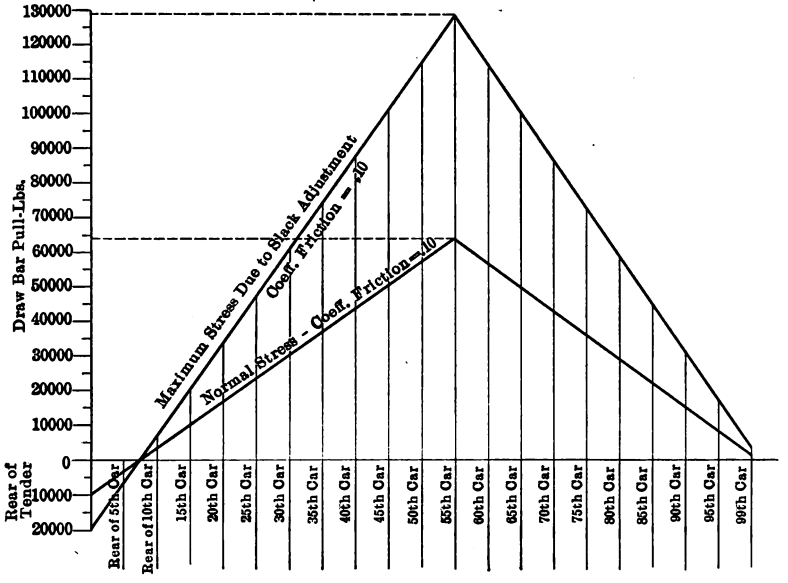


FIG. 47.—Stresses between cars in a train during braking.

in the forward part of the train is due to the greater proportion of braking power on the tender as compared with that of the other units.

The drawbar stresses above referred to are set up quite gradually during service application of the brake, and very suddenly and to an even greater extent, in emergency applications.

During service application the slack between the cars is taken up, when the train is said to be "bunched," and let out, when the the train is "stretched."

The application first taking effect upon the front part of the

train ordinarily permits the slack to become bunched and later when effective retarding force is reached on the rear section—because of its greater proportion with respect to the weight to be retarded—the train is stretched. This action at times occurs very quickly and causes jerks or sudden strains, which, combined with the stresses due to brake action, are frequently sufficient to damage draft rigging. A curve of calculated maximum stress due to slack adjustment is also shown in Fig. 47.

Problem 17.—With data for a K2 locomotive as given in Chap. II, find the stresses between the tender and first car and between the tenth and eleventh cars, with 60 loaded cars ahead and 60 empty behind, the weight of the loaded cars being 140,000 lb. and empty 40,000 lb. Take braking power as given in this chapter.

QUESTIONS ON CHAPTER VIII

1. What are the essential differences between the Straight Air and the Plain Automatic System? What caused the former to be dangerous?

2. What is the function of the High Speed Reducing Valve?

3. Define Adhesion. What causes it to vary in amount? What causes it to change so much when the wheels are locked?

4. With how much force should brake shoes be pressed against the wheels?

5. What is meant by "Braking power 110 per cent. light weight of car?" By "Equalized pressure in brake cylinder?" By "Factor of retardation?" By "Factor of retardation in per cent.?"

6. Explain the meaning: "Deceleration-Miles per hour per second." Define: "Normal Braking Ratio."

7. Draw a speed-distance curve similar to one shown in Fig. 41 and sketch a curve of adhesion and a brake-shoe friction curve showing where they would cross each other and explain the results occasioned by their crossing.

8. In Fig. 41 explain why the resistance curve is nearly horizontal for some distance.

9. In order to prevent "skidding," how is the brake cylinder pressure adjusted to take care of the increase in brake-shoe friction with the decrease in speed?

CHAPTER IX

COMBUSTION AND FUEL ECONOMY

PRINCIPLES OF COMBUSTION; AIR REQUIRED FOR BURNING COAL; AIR EXCESS; HORSE POWER TO PRODUCE DRAFT; PERFECT COMBUSTION CHAMBER; PROPERTIES OF PRINCIPAL COALS; HEAT LOSSES; LIQUID FUEL; MECHANICAL STOKERS; BRICK ARCH; POWDERED FUEL; DESIGN IN RESPECT TO FUEL ECONOMY; TEMPERATURE OF GASES IN BOILER TUBES; SPECIAL REVIEW PROBLEMS

During the year 1913, 534,466,580 short tons of coal were mined in the United States, about one-fifth of which was consumed by the locomotives in this country. The report of the Interstate Commerce Commission for the year ending June 30, 1912, shows that nearly $225,000,000 or 11.83 per cent. of all railroad operating expenses was expended for fuel, this representing nearly two-thirds of a million dollars a day for this item alone.[1]

Only 5 to 8 per cent. of the energy represented in the heat of combustion of every pound of coal burned in the fire-box of a locomotive is available as drawbar pull. While the overall efficiency is low, the locomotive compares favorably with the stationary power plant. For example, in a stationary boiler about 10 sq. ft. of heating surface is allowed for each boiler horse power, whereas in a locomotive 2 sq. ft. is the average value.

We are concerned with both *economy* and *efficiency* and a saving in the coal pile offers an opportunity for the best engineering skill. It is one of the vital problems in operation. The fireman, however, faces the immediate necessity of holding the steam pressure, and when that demand means pushing the boiler capacity to its limit, any theory about economy of combustion has little weight in the method followed to secure the necessary results. The engineer may design a locomotive with a view to burning a particular grade of fuel, but it is quite another problem to secure the best possible results from that fuel.

[1] For complete discussion of fuel expense, see "The Cost of Locomotive Operation," by G. R. Henderson.

Theory and practice are based on certain well-established principles, which must be thoroughly understood.

Combustion.—Carbon unites with oxygen, and the combustion may be complete or incomplete. *Complete* combustion results when carbon is supplied with all the oxygen with which it can unite. When complete combustion of carbon takes place, an inert, combustible gas called carbon dioxide is formed, represented by the chemical symbol CO_2.

If the oxygen supply for the carbon is deficient, *incomplete* combustion will take place. In this case a combustible gas, carbon monoxide, and represented by the symbol CO, is evolved.

When 1 lb. of pure carbon is burned completely to CO_2, 14,500 B.t.u. are given off. When 1 lb. of pure carbon is burned to CO, 4400 B.t.u. are evolved. Furthermore, when the carbon monoxide resulting from the incomplete combustion of 1 lb. of carbon is burned, the product of the combustion is carbon dioxide, and the heat evolved is 10,100 B.t.u. The same amount of heat is evolved, when a pound of carbon is completely burned, whether the combustion proceeds in one process or goes through two processes. If the combustion passes through two stages, however, and some of the carbon monoxide passes away unburned, the heat which it would furnish, at the rate of 10,100 B.t.u. per lb. of carbon, is lost.

The amount of CO_2 formed is always the same for the complete combustion of a given weight of carbon, no matter how much excess air is supplied. The volume of carbon dioxide compared with the total volumes of all the gases (CO_2, CO, hydrocarbons, air, nitrogen, etc.) leaving the furnace, becomes smaller as the amounts of the other gases increase.

The complete combustion of 1 lb. of hydrogen forms steam and about 62,000 B.t.u. are given off.

The oxygen necessary for the combustion of coal in a furnace comes from the air. Air is a mixture of oxygen and nitrogen in the proportion of 20.8 per cent. and 79.2 per cent. (by volume) respectively. The nitrogen of the air passes through the furnace without change. In order to be sure that sufficient air is provided, it is necessary to supply somewhat more than just enough air to carry in exactly the amount of oxygen for the combustion. About 40 to 50 per cent. excess air is commonly supplied, but in locomotive operation, the excess may be 150 per cent. and more.

When a shovelful of soft coal is thrown on a bed of hot in-

candescent fuel, the moisture and volatile gases are first driven off very rapidly, and in large volumes. The action is so rapid and the volumes of gas so large that frequently there is but little opportunity for proper mixture with air. Furthermore, some heat is necessary to warm up the coal and drive off the volatile matter and to heat up the incoming air, so that the process cools off the furnace.

With 50 per cent. air excess, it requires about 700,000 cu. ft. of air to supply the oxygen necessary to completely burn a ton of bituminous coal. A large locomotive hand fired, may consume at least 5000 lb. of coal an hour, resulting in the discharge of some 1,750,000 cu. ft., or 70 tons, of gases an hour—over a ton a minute, and all this gas is heated to 2000°F. or more. Either choking off this supply or, on the other hand, allowing it (because of poor firing) to pass through pockets or holes in the fuel bed, thus preventing intimate mixture of the gases at the higher temperatures necessary for complete combustion, are directly opposed to conditions of economical operation. One pound of the air entering at atmospheric pressure and 32°F. occupies 12.37 cu. ft.; at 1000° occupies 37.0 cu. ft.; at 2000°, 62 cu. ft., so that the volume entering the flues is from four to five times the volume of the same weight of gas entering the furnace.

Air for Combustion.—The atomic weights for the more common elements are: Hydrogen, 1; oxygen, 16; carbon, 12; sulphur, 32. Completely burning 12 lb. of carbon to CO_2, requires $16 \times 2 = 32$ lb. of oxygen giving $32 + 12 = 44$ lb. of CO_2; hence, each pound of C requires $\frac{32}{12} = 2.66$ lb. of O. If it receives but half of this amount of O, CO is formed and $12 + 16 = 28$ or $\frac{16}{12} = 1.33$ lb. of O to 1 of C. In the burning of hydrogen to water, the quantities by weight give $H_2 + O = H_2O$, $1 \times 2 + 16 = 18$ or 2 lb. of H + 16 lb. of O produce 18 lb. of water and each pound of H requires 8 lb. of O for its complete combustion.

Expressing the above by formula, the oxygen required for complete combustion = 2.66C + 8H. But the O in the fuel will unite with the H, leaving to be actually supplied,

$$2.66C + 8\left(H - \frac{O}{8}\right)$$

Since 4.25 lb. of air are required to supply 1 lb. of O, we may write:

Air for combustion (without excess air) =

$$4.25\left[2.66C + 8\left(\mathrm{H} - \frac{\mathrm{O}}{8}\right)\right] = 11.3C + 34\left(\mathrm{H} - \frac{\mathrm{O}}{8}\right) \quad (73)$$

Expressed in terms of cubic feet of air =

$$158C + 473\left(\mathrm{H} - \frac{\mathrm{O}}{8}\right)$$

$$\text{Pounds of air per lb. carbon} = \frac{3.032\mathrm{N}}{\mathrm{CO_2} + \mathrm{CO}} = \mathrm{Wa} \quad (74)$$

in which N, CO_2 and CO are the percentages of dry gas obtained by analysis.

Pounds air per pound of coal = pounds air per lb. C × per cent. C in the coal. Ratio air supply to that theoretically required for complete combustion = $\frac{\mathrm{N}}{\mathrm{N} - 3.782\mathrm{O}}$.

Observations show that incomplete combustion is common even when large amounts of air are involved. As high as 20 per cent. of the heat in the coal has been found to be carried away by the sparks during rapid combustion rates and the total loss from unburned combustible gases may be over 20 per cent. in bituminous coal. (See Table IX, this chapter.) The problem in design is to allow sufficient air openings, properly placed, to permit air to pass up freely from the sides and back of ash pan. It has been found to be good practice to make the openings into the ash pan at least 15 per cent. of the area of the grate, this giving a vacuum of less than 1 in. of water under the most rapid evaporating rates.

A careful study should be made of Table VI. Note the much larger volume of gaseous products per pound when burning C to CO_2 over that of the incomplete combustion to CO, and compare this with the air for hydrogen.

Problem 18.—If 5000 lb. of carbon are burned per hour on a grate with 100 per cent. air excess, how much more heat will be required by the air than if the fuel had burned with 50 per cent. air excess? (Hint: take the specific heat of air from a text-book on heat and use other values from Table VI.)

Problem 19.—Check temperatures for air supply in Table VI, for 50 per cent. and for 100 per cent. air excess, assuming 1 lb. of carbon contains 14,600 B.t.u. (Hint: write an equation for the heat taken up by the elevation of temperature in question, solving for the elevation of temperature.)

TABLE VI.—OXYGEN AND AIR REQUIRED FOR THE COMBUSTION OF CARBON, HYDROGEN, ETC.

	Chemical reaction	Lb. O per lb. fuel	Lb. N. = 3.30 × O	Air per lb. = 4.32 × O	Gaseous product per lb.
Carbon to CO_2	$C + 2O = CO_2$	2⅔	8.85	11.52	12.52
Carbon to CO	$C + O = CO$	1⅓	4.43	5.76	6.76
Carbon monoxide to CO_2	$CO + O = CO_2$	⅔	1.90	2.47	3.47
Hydrogen to H_2O	$H_2 + O = H_2O$	8	26.56	34.56	35.56
[1]Methane, CH_4 to CO_2 and H_2O	$CH_4 + 2O = CO_2 + 2H_2O$	4	13.28	17.28	18.28
Sulphur to SO_2	$S + 2O = SO_2$	1	3.33	4.32	5.32

Temperature of the Fire—Carbon Burned to CO_2 with Excess of Air

Air-supply above 11.52 lb., per cent.	35	50	75	100	150	200
Air per lb. C, lb.	14.40	17.28	20.16	23.04	28.80	34.56
Air + C = gas, lb.	15.40	18.28	21.16	24.04	29.80	35.56
Elevation of temperature of fire	3950°	3328°	2875°	2530°	2041°	1711°

[1] One of the compounds which make up "Volatile Matter" in coal.

Air Excess.—Analyzing a sample of the escaping gases by an Orsat apparatus, it is possible to determine the air in excess and the oxygen actually used.

Suppose that a certain analysis of flue gases gave

O = 10.2 per cent.
CO = 0.0 per cent.
CO_2 = 8.4 per cent.
N = 81.4 per cent.

$$\text{Nitrogen} \times \frac{\text{oxygen}}{\text{nitrogen}} = \text{N} \times \frac{21.8}{79.2}$$

= total oxygen in air = N × 0.274. Then O_2 in excess = O_2 in air − O_2 in excess.

$$\text{N} \times 0.274 =$$
$$81.4 \times 0.274 = 22.4 = \text{total } O_2 \text{ in air.}$$
$$10.2 = O_2 \text{ in air excess.}$$
$$12.2 = O_2 \text{ used.}$$
$$\frac{10.2}{12.2} = 83 \text{ per cent. air excess.}$$

Problem 20.—Analysis, O = 12.6; CO_2 = 6.0; CO = 0.1; N = 81.3. Find air in excess.

Air Supplied per Pound of Combustible.—Take the analysis of a sample of Penn Gas coal:

ULTIMATE ANALYSIS—PER CENT.

	Dry coal as base	Combustible as base
Carbon	79.2	84.60
Hydrogen	5.1	5.45
Nitrogen	1.5	1.61
Sulphur	1.6	1.71
Ash	6.4	.....
Oxygen by difference	6.2	6.63
Total	100.0	100.00

$$\text{Column 2} = \frac{\text{Column 1}}{100 - 6.4\ (\text{ash})} \times 100$$

The smoke-box gases contained by volume under heavy firing in an E6s (Atlantic type) locomotive, gave:

CO_2 = 14.0 per cent.
CO = 2.2 per cent.
O = 0.2 per cent.
N = 83.6 per cent.

100.0 per cent.

$$W_g = \frac{11CO_2 + 8\ O + 7(CO \times N)}{3(CO_2 + CO)} \times \left\{ \begin{array}{c} \text{carbon per pound} \\ \text{combustible} \end{array} \right\} \quad (75)$$

$$W_a = \frac{3.032N}{CO_2 + CO} \times \left\{ \begin{array}{c} \text{carbon per pound} \\ \text{combustible} \end{array} \right\} \quad (76)$$

W_g = wt. (in pounds) of gas per pound combustible.
W_a = wt. (in pounds) of air per pound combustible.

Problem 21.—For Penn Gas coal, using above analysis for coal and gas, find values of W_g and W_a.

Horse Power to Produce Draft.—The calculation for determining the power to draw the combustible gases through the flues and under the diaphragm, and to deliver them to the stack, reduces to the following expression:

$$\text{Horse power} = \frac{PAV}{33{,}000} \text{ or approximately } = \frac{PW}{93.5}$$

in which,

P = velocity pressure, or draft, in inches of water against which the gases are pulled. This pressure, measured by U tubes, is usually taken in front of tube sheet and near the stack.

A = total area of tubes (and flues), square feet.

V = average velocity of gases in feet per minute = $\frac{FW}{60A}C$.

F = cubic feet air required per pound of fuel = approximately 185 cu. ft. per lb. soft coal burned.

W = pounds fuel burned per hour.

C = factor for increase of volume of gas due to increase from outside temperature (about 60°F.) to front-end temperature of 700°F. (= approximately 2.2).

hp. = $\frac{144PFW}{198{,}000}C$. The values of C and F being about the same as noted for average conditions in practice, we may write:

$$\text{(approx.) } H.P. = \frac{PW}{93.5} \tag{77}$$

Problem 22.—Fuel per hour = 7300 lb., draft in front end = 15 in. water, draft in front of tube sheet = 8 in. water. Area flues = 9.19 sq. ft. Find the horse power (*a*) to draw the gases through the tubes and (*b*) to draw them through and to discharge them at the front end.

Figure 48 gives the average results of draft, for different combustion rates, on a 4–6–2 type locomotive as observed on the test plant at Altoona. It shows a nearly uniform increase in draft, at each place where observations were taken, for the increase in dry coal fired per hour. It may be noted that the loss in draft between the front and the diaphragm is about 50 per cent. of the total draft produced by the steam exhaust. In the tests on this locomotive the circular nozzle originally applied was displaced by a rectangular one measuring $4\frac{5}{16} \times 6\frac{5}{8}$ in. and this in turn was modified to an elliptical nozzle having the same area as the rectangular nozzle, the elliptical nozzle giving the best draft conditions obtained in the series of tests. Following these tests, there was developed at the test plant a nozzle having four internal projections, the results of which are proving satisfactory.

Recent tests on powdered fuel for a locomotive show a lower back pressure in the cylinders than when run of mine coal is used, indicating less power required for draft when using powdered fuel. (Railway Age Gazette, Mechanical Edition, May, 1915.)

The Perfect Smokeless Combustion Chamber.—Reduced to its simplest statement, the conditions necessary to produce complete combustion are twofold: (1) The fuel must be intimately mixed with the required amount of air. (2) The temperature necessary

for complete combustion must be maintained for a sufficiently long period of time to obtain the required chemical combination.

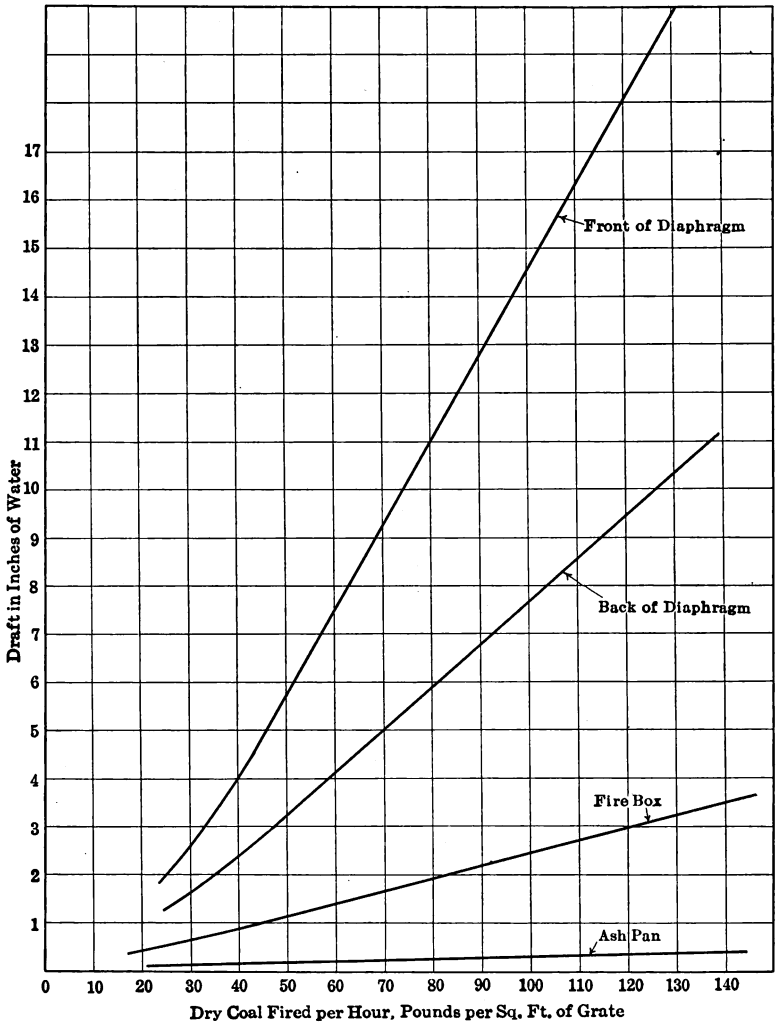


FIG. 48.—Draft in terms of dry coal fired. (Results on a 4-4-2-type.)

Above it will be noted that the increase in draft for an increase in the rate of dry coal fired is uniform, and that but half of the total draft is effective back of the diaphragm, as shown by examination of the two top inclined lines.

The fireman, however, is the most important factor in securing the best conditions of combustion.

In the ideal furnace, the fuel composed of carbon, various hy-

drocarbon gases and perhaps sulphur is burned in air. Each atom of the fuel finds and seizes upon the oxygen atoms with which it will combine. When combined, the resulting gases pass out of the furnace carrying the heat of combustion. The student is advised to study for himself the simple principles of combustion by aid of a common coal oil or kerosene lamp.

The upper part of the chimney of the lamp may be considered to be the stack. The larger part of the chimney is, in analogy, the combustion chamber. The small space just above the wick is the fuel bed and the air openings below are the openings through the grates for air supply.

1. Remove the chimney, open the hood and light the wick, which should be evenly trimmed. If turned low, it does not smoke. Close the hood and the flame smokes, because the hood has cut off the air about the flame, thus illustrating a furnace with insufficient air where most needed, a condition which occurs when too much fuel (for the required air supply) is on the grate.

2. Without changing the condition of the flame place chimney on the lamp. It does not smoke and the flame brightens, showing better combustion because the increased draft draws more fresh air about the wick just where needed.

3. Chill the flame by means of a piece of iron held in it, thus cooling the gases before combustion is complete, illustrating the condition when too much cold air strikes the flame.

4. When burning brightly with the chimney on, shut off the air supply by wrapping a cloth about the hood, closing of the required air supply, and the flame smokes, as it does when the dampers in the fire-box are closed too much.

5. Raise the chimney from its seat; the flame is chilled and the lamp smokes. Open wide the fire door of the locomotive and the same thing happens, the air being chilled before combustion has been completed. In practice, the fire door must frequently be "cracked" enough to add more oxygen than can be supplied through the grates.

These observations teach us that the essential conditions for preventing smoke and securing better fuel economy are:

1. That the fuel be supplied in small quantities and just enough air be passed up through the grates to burn it.

2. That the gases be distilled from the coal at a uniform rate. (The fireman should hold a uniform bed of coals.)

3. That the air be heated by passing through the bed of hot coals.

4. That the volatile gases given off shall mix with the fresh air supply so that each particle of the carbon elements gets its necessary supply of air.

Smoke.—It is understood that smoke comes chiefly in the burning of bituminous coal. The losses in the black smoke itself are not large, scarcely ever exceeding 1 per cent. of the heat in the fuel. Smokeless combustion is an *indication* of complete combustion of the fuel, but a furnace working with a minimum air supply may give out dense clouds of smoke and still give a higher evaporation than one made smokeless by a large excess of air. Consequently "smokeless combustion" may only indicate certain conditions in the fuel bed, and should not be confused with the question of fuel economy.

Smoke is not necessarily formed under conditions of low temperature. There must be a temperature of at least 1800°F. in the fire-box before it is possible for black smoke to be formed, for the reason that black smoke is the carbon of hydrocarbon gases evolved from the coal; these hydrocarbon gases are given off at temperatures ranging from 400 to 1000°F. and do not separate into carbon and hydrogen until a temperature of at least 1800° is reached, and until that temperature is reached and a separation brought about, black smoke is not visible. Therefore, the formation of *black smoke is not due primarily to a low fire-box temperature,* but rather to the production and heating of the gases to the point at which they separate into hydrogen and carbon and then failing to have enough air supplying oxygen to burn them at that time and at the proper temperature. Any part of the carbon that goes away unsupplied with oxygen goes away as free carbon, which is jet black and called black smoke. It is not the reduced fire-box temperature which causes black smoke so much as the presence in the fire-box of an amount of fuel in excess of the oxygen supplied by the incoming air to burn that fuel.

Professor Breckenridge sums up the conditions as follows: "If there is a deficient air supply, part of the carbon atoms will not find enough oxygen atoms with which to combine and there will be a considerable part of the escaping gases leaving the chimney as carbon monoxide instead of burning to carbon dioxide. For each pound of carbon burned only to carbon monoxide there will be a loss of approximately 10,100 heat units, and this consti-

tutes the great source of loss so frequently referred to as the loss due to incomplete combustion. This loss may readily amount to 5 per cent. of the total heat in the coal. The density of the accompanying smoke may or may not be an indication of the proportion, though the loss due to carbon monoxide in perfectly smokeless chimney gases in practice will usually not exceed 0.05 of 1 per cent. Smokelessness is a relatively safe indication that the total heat has been liberated. Unfortunately it gives no indication of the degree of efficiency with which the heat is being utilized. The problem from the standpoint of the operator demands smokelessness with a minimum air supply."

In 1912 smoke tests were made on the locomotive testing plant of the Pennsylvania Railroad and later were reported before the American Railway Master Mechanics' Association, with recom-

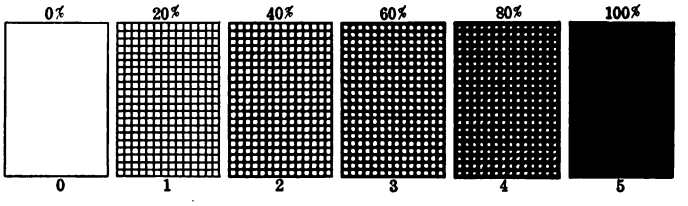


FIG. 49.—The Ringelmann scale for grading the density of smoke.

mendations covering the application of steam-air jets, quick-action blower valves, etc. Mr. E. W. Pratt of the C. & N. W. R. R. reports[1] that since that time practically every locomotive operating in the city of Chicago has been equipped with such apparatus, and it has been conclusively proven that soft-coal-burning locomotives may thereby be kept comparatively free from smoke if the engine crew be given, and observe, proper instructions at all times. The reduction in density of railroad smoke is seen from the following figures: In 1912 the per cent. smoke density was 10.74 per cent., while in 1913 it was 6.06 per cent., these percentages being based on the Ringelmann chart, shown in Fig. 49, for determining smoke density, and the observations being made by inspectors who take smoke readings for a total of 2 hr. each day.

The use of powdered fuel on locomotives suggests one of the attractive methods for the material reduction or entire elimination of the smoke nuisance on locomotives.

[1] Railway Age Gazette, May 21, 1915, included in report before International Railway Fuel Association.

COAL

The United States Geological Survey suggests that coal may be graded on the basis of the ultimate analysis, which establishes the ratio of hydrogen to carbon, and is the quotient of the percentage of C divided by the percentage of H, or $R = \frac{C}{H}$.

TABLE VII.—CLASSIFICATION OF COAL

Class	Carbon hydrogen ratios, $\frac{C}{H}$
Anthracite	over 30 to 30; 30 to 26
Semi-anthracite	26 to 23
Semi-bituminous	23 to 20
Bituminous	20 to 17 — gas coal
	17 to 14.4 — gas coal
	14.4 to 12.5 — coking coal
	12.5 to 11.2 — coking coal
Lignite	11.2 to 9.3 (black or sub-bituminous brown)

Anthracite coal burns more slowly than bituminous, and, consequently, a larger grate area has to be provided in order that sufficient coal may be burned to give the required amount of steam. In other words, means must be provided to make a hard-coal-burning locomotive of given proportions consume as much coal per hour as a bituminous burner of the same proportions; and no better way has been found than by designing this kind of a locomotive with a large fire-box and a liberal grate area. Anthracite coal has to be fired to suit the size of the lumps used. If the coal is in large lumps, a heavy fire must be carried, because the lumps lie so open that the air would pass too freely through the fire if it were light.

TABLE VIIa.—TYPICAL AMERICAN COALS (U. S. Geological Survey)

	Anthracite		Semi-bituminous		Bituminous				Lignite	
	Lehigh mine run	Scranton culm	Pocohontas mine run	Arkansas half lump, half slack	Big Muddy, Carterville, Ill., mine run	Jamison, Penn., lump	Straight Creek, Ky., mine run	St. Claire County, Ill., slack	Tesla, Cal., mine run	Gallup, New Mexico, mine run
Proximate analysis (per cent.):										
Water	1.97	2.08	0.65	1.28	8.61	3.15	1.92	5.31	18.51	10.86
Volatile matter	4.35	7.27	18.80	12.82	32.40	30.27	36.56	34.29	35.33	35.14
Fixed carbon	86.49	74.32	75.92	73.69	51.33	56.17	57.08	36.24	30.67	46.90
Ash	7.19	16.33	4.63	12.21	7.66	10.41	4.44	24.16	15.49	7.10
Sulphur	0.64	0.77	0.57	2.01	1.65	1.26	1.24	4.30	3.05	0.64
Ultimate analysis (per cent.):										
Carbon	85.66	75.21	85.91	77.29	68.14	74.33	78.31	54.06	47.34	64.34
Hydrogen	2.78	2.81	4.58	3.74	5.38	4.96	5.36	4.57	5.93	5.73
Nitrogen	0.77	0.80	1.07	1.39	1.34	1.43	1.85	0.78	0.66	1.05
Oxygen	2.87	4.08	3.24	3.36	15.83	7.61	8.80	12.13	27.53	21.14
Calorific value: B.t.u.'s per pound fuel:										
Calorimeter	.....	12,472	15,190	13,406	12,236	13,406	14,319	9,848	8,525	11,435
Dulong's formula	13,963	12,395	15,104	13,831	12,082	13,371	14,081	9,929	8,550	11,299

SEMI-BITUMINOUS COALS

Class	State	Heating value, B.t.u.'s per lb. fuel	Percentage						Condition of coal	Part of state	Name of coal or coal bed
			Carbon	Hydrogen	Sulphur	Volatile	Ash	Moisture			
Semi-bituminous	Maryland	13,680	77.60	3.85	1.55	4.24	12.76	.....	Run of mine	N. W.	Lower Kittaning.
	Pennsylvania	14,659	82.88	4.13	1.51	3.99	7.49	.....	Run of mine	S. W.	
	Pennsylvania	15,609	58.35	.....	.....	35.66	4.71	1.29	.....	.....	.....
	Pennsylvania	14,752	69.90	.....	1.44	21.1	8.25	0.75	.....	.....	.....
	West Virginia	14,180	80.34	4.00	0.49	4.16	11.01	.....	Run of mine	S.	Pocahontas.
	West Virginia	15,033	84.16	4.66	0.65	5.39	5.14	.....	Run of mine	S.	New River.

Henderson gives the following table:

TABLE VIII.—RATIO OF HEATING SURFACE TO GRATE AREA

Fuel	Passenger		Freight	
	Simple	Compound	Simple	Compound
Free-burning bituminous	65 to 90	75 to 95	70 to 85	65 to 85
Average bituminous	50 to 65	60 to 75	45 to 70	50 to 65
Slow-burning bituminous	40 to 50	35 to 60	35 to 45	45 to 50
Bituminous slack and free-burning anthracite	35 to 40	30 to 35	30 to 35	40 to 45
Low-grade bituminous, lignite and slow-burning anthracite	28 to 35	24 to 30	25 to 30	30 to 40

To find the coal consumed in terms of the per cent. of theoretical tractive effort and speed, see page 224. The effect of cut-off on coal consumption is referred to on page 214, Fig. 97.

The total heat of combustion may be computed from the ultimate analysis by Dulong's formula:

$$\text{B.t.u.} = 14{,}600\text{C} + 62{,}100\left(\text{H} - \frac{\text{O}}{8}\right) + 4000\text{S}$$

where

B.t.u. = heating value per pound dry fuel.

C, H, O and S = parts of carbon, hydrogen, oxygen and sulphur.

The heat content may be readily determined by a standard coal calorimeter (as the Mahler's Bomb).[1]

Proximate analysis gives relative values for the moisture, volatile (combustible) matter, carbon and ash, named in the order in which their percentage is determined by such analysis. The volatile matter and carbon are the combustibles from which heat is supplied. The sulphur, an undesirable constituent, is separately determined.

The moisture referred to is the loosely held water which is driven off by heating a finely ground sample for 1 hr. at 105°C. The amount, thus determined, may be less than 1 per cent. or it may exceed 10 per cent., the percentage also being higher when coal is first mined than after it has been air dried.

[1] For details of methods of testing and general properties of fuel the reader is advised to consult the excellent treatise on "Coal" by Prof. E. E. Somermeier, of Ohio State University, published by McGraw-Hill Book Co., Inc.

The volatile matter consists essentially of a part of the sulphur, nitrogen, hydrocarbons of unknown and varying composition and any combined water in the coal. The amount of volatile matter is determined by heating a one-gram sample of fine coal in a platinum crucible (with a close-fitting cover) for 7 min. over a Bunsen flame. The crucible and residue are cooled and weighed, the loss in weight in per cent. less the weight of moisture in the sample in per cent. is the volatile matter in per cent. In anthracite coals, the amount is small, from 1 to 8 per cent., while in bituminous, semi-bituminous and lignites it may run as high as 45 per cent.

The ash is a non-combustible residue, the weight of which is determined after burning off the fixed carbon in a muffle furnace and the fixed carbon is the difference between 100 and the sum of the moisture, ash and volatile matter.

Heat Losses.—One pound of the average Penn Gas (bituminous) coal gives off about 14,300 B.t.u. when burned to CO_2. About 60 or 70 per cent. of this heat is present in the steam formed, the remaining part supplying the following losses:

A. There is a (controllable) loss of about 5 per cent. due to unconsumed fuel passing through the grate. This loss varies with the skill of the fireman and the method of handling the coal on the grate.

B. The (controllable) loss due to the incomplete combustion of the carbon which burns to carbon monoxide and this may amount to 10 per cent. Sufficient air supply and its proper mixing with the combustible gases reduces this loss.

C. The (unavoidable) loss in evaporating the moisture in the coal is usually less than 1 per cent., but may reach 4 per cent.

D. There is a further (unavoidable) loss due to the moisture formed in burning hydrogen, amounting to about 4 per cent.

E. It is estimated that from 2 to 5 per cent. is radiated from the furnace and boiler and this is unavoidable.

F. Another large loss is due to the heat carried away in the dry chimney gases. This loss may be over 20 per cent. but by a careful regulation of air supply may be reduced to 10 per cent.

G. Unconsumed fuel passes out of the stack in the form of carbon particles and little tar globules. This loss is difficult to accurately determine. It has been found in an E6s to be as

high as 11.5 per cent., this amount occurring when the dry coal fired per square foot of grate slightly exceeds 120 lb. and became less as the combustion rate was reduced. The loss due to black smoke alone, which is largely soot, is probably not over 2 per cent.

The last two losses mentioned (*F* and *G*), may, in part, be avoided, first by means of proper design and second by operating in a way favorable to the conditions under which the fuel is burned.

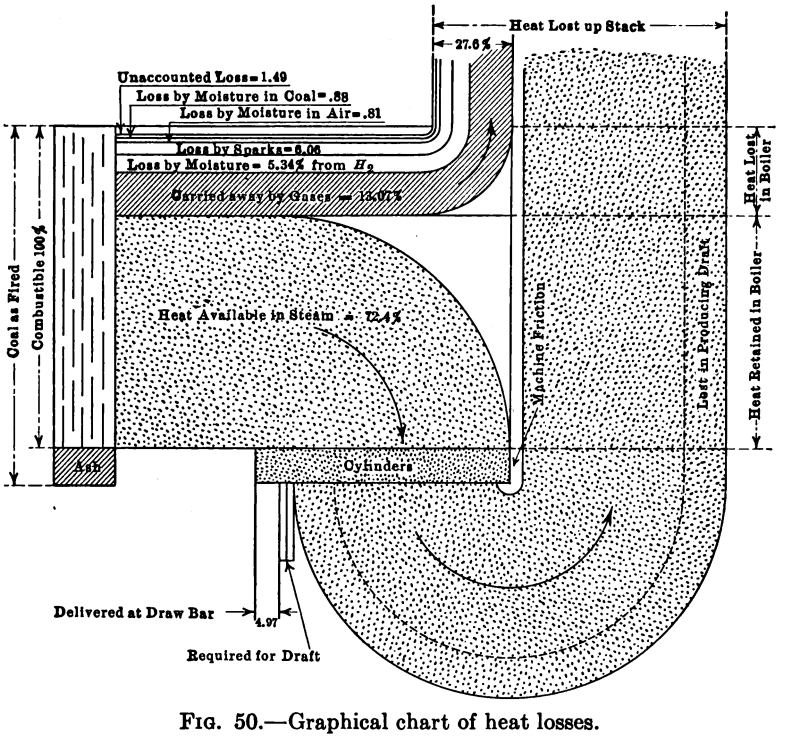


FIG. 50.—Graphical chart of heat losses.

The practical application of the foregoing is brought out in the following heat balance, to which are added methods for calculation of results. This is a fair average condition but giving less percentage available at the drawbar than given by some other heat balances which have been worked out. The final results are plotted in Fig. 50, this particular form of diagram being suggested by the author as especially adapted to illustrate the heat losses in a locomotive.

DATA FOR HEAT BALANCE

Locomotive Type 2-8-0 Saturated Steam

ULTIMATE ANALYSIS OF COAL

Carbon	64.85
Hydrogen	5.45
Nitrogen	1.16
Sulphur	3.37
Ash	11.35
Oxygen by difference	13.82
	100.00

PROXIMATE ANALYSIS

Fixed carbon	Volatile matter	Moisture	Ash	Sulphur, separately determined
40.7	40.92	7.56	11.35	3.37

B.t.u. per lb. dry fuel = 11,835
B.t.u. per lb. dry combustible = 13,337

ANALYSIS OF FLUE GASES

O	CO	CO_2	N
5.8	0.0	12.4	81.8

Total coal as fired, lbs.	Conditions	Total sparks, lbs.	Moisture in sparks	Dry sparks, lbs.	Dry sparks discharged, per cent. of total dry coal fired
8493	80–35–F	423	4.7	403	5.32

Equivalent evaporation from and at 212°F. per lb. of dry fuel = 8.81
Temperature of feed water..... 64.2
Temperature of air in test room..... { Dry bulb, 55° / Wet bulb, 62°
Temperature of flue gas, 550°
Temperature of fire-box end flues, 1810°

TABLE IX.—HEAT BALANCE ON BASIS OF COMBUSTIBLE
(SLIDE RULE CALCULATIONS)

	B.t.u.	Per cent. combustible
Total heat value of 1 lb. of dry coal	11,835	
Total heat value of 1 lb. of combustible	13,337	
1. Heat absorbed by the boiler-evaporation from and at 212° per lb. of combustible × 970.4.	9,650	72.40
2. Loss due to moisture in coal = per cent. of moisture referred to combustible ÷ 100 × total heat above test-room temperature of 1 lb. steam.	119	0.88
3. Loss due to moisture formed by the burning of hydrogen = per cent. of hydrogen to combustible ÷ 100 × 9 × total heat above test temperature in 1 lb. steam.	714	5.34
4. Loss due to heat carried away in dry chimney gases = weight of gas per pound of combustible × 0.24 × $(T - t)$. T = flue gas temperature, t = room temperature.	1,743	13.07
5. Loss due to incomplete combustion of carbon = $\frac{CO}{CO_2 + CO} \times \frac{\text{per cent. C in combustible}}{100} \times$ 10,150.	0	0.0
6*a*. Loss in sparks	810	6.06
6*b*. Loss in moisture in air entering	108	0.81
6*c*. Other losses	195	1.49

Bulletin 18, Test Department, Pennsylvania Railroad, gives the following results of a heat balance under different rates of combustion with coal of average heat value of 14,616 B.t.u. per lb. dry coal. When the combustion rate has exceeded 6000

TABLE IX*a*.—HEAT BALANCE BASED ON DRY COAL

Dry coal fired per hour, pounds	Heat absorbed by boiler, per cent.	Heat lost due to							Total, per cent.
		Evap. of moisture in coal, per cent.	Steam formed by burning hydrogen, per cent.	Heat in dry gases, per cent.	Carbon monoxide, per cent.	Heating moisture in air, per cent.	Sparks, per cent.	Radiation and unaccounted for, per cent.	
2539	75.13	0.18	4.05	10.58	0	1.05	4.79	4.18	99.96
2596	77.88	0.18	4.13	12.15	0.56	0.74	3.45	0.93	100.01
2995	70.47	0.17	4.01	13.95	0	0.63	8.50	2.27	100.00
3738	68.95	0.17	4.08	13.42	0	0.75	5.19	7.53	99.90
3855	63.99	0.18	4.06	15.81	0	1.23	5.49	8.22	100.00
4300	68.29	0.21	4.13	16.33	0	0.74	8.29	1.99	99.98
5257	59.08	0.18	4.13	12.55	0.38	0.62	11.89	11.07	99.90
5728	62.11	0.17	4.19	17.55	1.58	0.75	11.68	1.87	99.90
8412	47.8	0.18	4.28	13.9	10.2	1.04	21.5	1.10	98.80

lb. coal per hour, the loss from the presence of carbon monoxide and from sparks rapidly increases. The boiler holds up well in efficiency until the highest rate of coal fired as given in Table IX*a* is reached. This table should be studied in the light of the causes of the various losses, previously discussed.

INFLUENCE OF LOAD AND SPEED ON FUEL CONSUMPTION.—In his book on "Locomotive Operation," Henderson gives a study of the effect of load and speed on fuel consumption, to which the student is referred for a more extended study. This same problem has been investigated with care at the Altoona test plant for many locomotives, the results of which will be found in recent Transactions of the American Railway Master Mechanics Association. It has also been shown that a saving in coal from 20 to 30 per cent. may be expected in the same simple locomotive using superheated steam (about 250° superheat) over that obtained when working with saturated steam. In a series of tests on a E3sd (Atlantic) type locomotive, the following relation was established: $E = 13.2 - (0.057)C$, where E = equivalent evaporation per pound dry coal and C = dry coal per hour, pounds per square foot of grate, thus showing a straight line relation for decrease in evaporation with decrease in coal burned.

LIQUID FUEL

During the past few years there has been a marked increase in the use of fuel oil for locomotives, especially on the western roads. In an oil-burning locomotive, the burner should be placed in the front end of the draft pan and directed toward the rear in such a manner that the draft is forced to reverse the direction of the flame before it passes to the flues. The oil fuel is forced by steam pressure through a nozzle giving a uniform distribution of heat over the entire fire-box. The usual form of nozzle consists of a rectangular casting containing two longitudinal passages separated from each other by a horizontal partition. The oil flows through the upper passage to an opening in the face of the burner so arranged that it allows the oil to drool down on to the flat steam jet which issues from a similar but much smaller opening. The steam sprays the oil into fine particles and forces it back to the point of combustion.

The following are the chief advantages of oil fuel: The ease with which it can be controlled by adjusting a valve; air holes

and dead spots are not present; a constant temperature can be maintained by the adjustment of the regulating valve; complete combustion; no ashes and the absence of sparks and cinders. There being no necessity to open the fire door for the introduction of fuel, there is no reduction of pressure, chilling of flues, sheets, and leaking due to this cause. The absence of sulphur in the fuel makes its action on the boiler-plate less destructive than when using coal as fuel.

Coal requires more time for decomposition and the elimination of the products and supporters of combustion; compared with oil the combustion of coal is slow and requires aid by strong draft. Oil, having no ash or refuse when properly burned, requires much less space for combustion, for the reason that, being a liquid and the compound of gases being highly inflammable when united in proper proportions, it gives off its heat rapidly, and at the point of ignition is all ready for combustion. By burning oil with a clear white fire it is free from smoke, dust, and soot; also in locomotives the exhaust nozzles can be made the maximum size, because a strong blast is not required for combustion. This is an item to be considered, as the power of the engine is increased by the reduction of back pressure against the piston.

The heat in a pound of crude petroleum is about 20,000 B.t.u., and has given under test an evaporation of over 18.5 lb. of water. The success of the oil burner depends upon the fireman and he should be properly educated and directed.

LOCOMOTIVE MECHANICAL STOKERS

Many locomotive stokers have been applied to different classes of locomotives. It has been shown that this additional equipment gives: (1) Increased tonnage with existing locomotives, (2) possibilities of burning low-grade fuel, (3) a more uniform fuel bed, (4) ability to handle successfully the fuel under the highest rates of combustion, and (5) the construction of locomotives which may use more coal than can be properly handled by hand firing. There are two general types in use, the *underfeed* and the *overfeed* (of scatter) types, the "Crawford" stoker being one of the former and the "Street" being one of the latter types. To these has been added a third, namely a *chain grate* type represented in the Ayers stoker. Among the overfeed types are the Hanna, Standard, Gee, Kincaid, Elvin and Raite stokers.

Crawford Stoker.—All the mechanism of the stoker, including the conveyor from the tender, is operated by a single steam cylinder. The coal is fed from the tender through an opening in the floor and is then handled by a feeder and breaker sliding backward and forward in the direction of the center line of the tender. This is not a crusher, but has a limited opening, and is provided to prevent large lumps having access to the conveyor beneath. Inside of the conveyor are placed rake fingers operated from a bar above and extending nearly to the bottom of the trough, and having such a movement as to drag over the coal in moving backward and scraping the coal toward the locomotive in its forward movement. From the conveyor the coal is led to a point directly beneath the deck plate of the locomotive back of the boiler and dropped into a space where it is pushed by plungers under the mud ring and into two troughs longitudinal to the fire-box and below the surface of the grates.

In the bottom of the troughs are additional feeders for conveying the coal to the middle or forward portion of the grate. The plunger or feeder first taking the coal feeds it principally to the back end of the grate, and the additional feeders carry it forward. The arrangement of these feeders is such that their number can be increased so as to get the desired distribution of coal. The operation of the steam cylinder is controlled by a small valve placed in the cab.

The Street stoker (Fig. 51) is designed to fire heavy locomotives with screened or crushed coal, distributed by steam blast through additional openings through the back-head of boiler and scattered over all parts of the grate area continuously, in contrast to the method of distributing supply of coal to different parts of the grate area intermittently.

The Street stoker consists essentially of: A screw conveyor, placed under the floor of the tender, for carrying the coal from the tender to the locomotive in such quantity as may be required, as regulated by size of coal opening used over the screen and the speed of stoker as regulated at any one of the seven notches on the variable-speed governor lever quadrant. An elevator in the cab on back-head of boiler, for receiving the coal and elevating it to the discharge pipe or distributing center. A regulating system which re-screens the coal and furnishes proper quantities for each firing zone and a steam jet distributing system for spreading the coal over the grate area as may be required to

maintain an even fire and maximum steam pressure under the direction of the fireman.

The screen over the screw conveyor below the coal opening in bottom of locomotive tender has 2½ in. square openings and is intended to handle coal of size commonly known as "nut run," that is, a nut, pea and slack mixture.

The automatic movement of the screen under the coal opening

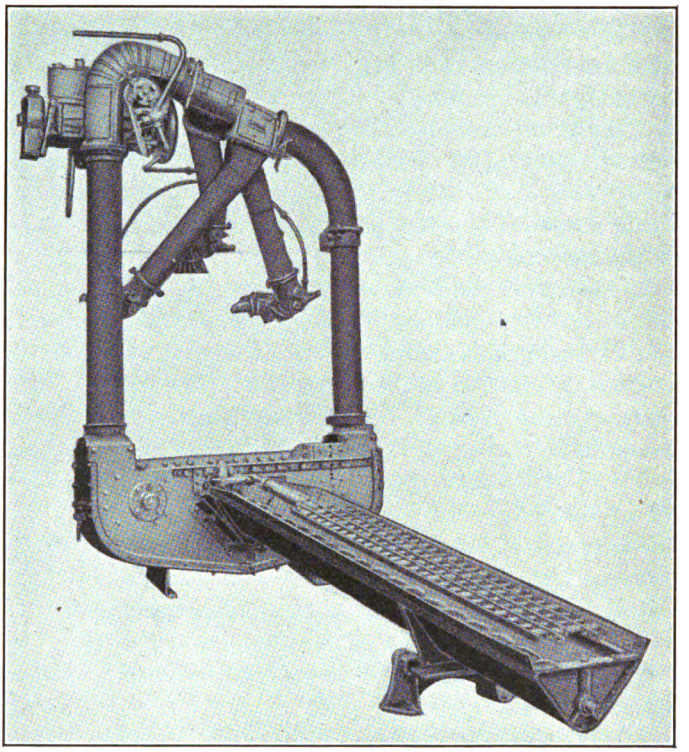


FIG. 51.—The Street locomotive stoker.

will cause the coal to enter conveyor trough, the helicoid screw will convey coal to elevator hopper and elevator buckets will elevate the coal to the discharge pipe or distribution center. The finest coal is then screened in proper proportion as determined by size of openings the fireman turns to register in discharge pipe screen.

The fine coal goes down center distributor pipe and is fired by the steam jet from nozzle in center elbow into the fire-box against

the center distributor which deflects the fine coal downward and spreads same over the central back part of fire-box.

The coarser coal is carried past the discharge pipe screen by the elevator buckets and falls through large opening in right-hand part of discharge pipe and is divided equally, or unequally as may be determined by the fireman, between the two side distributor pipes. The coarse coal is fired by the steam jets in right and left elbows into fire-box and is spread by right and left distributors over the main right and left fire-box areas. The distribution overlaps as between the three areas or zones fired from the three elbows. This overlapping insures ample coal being supplied to center of fire-box in heaviest combustion area.

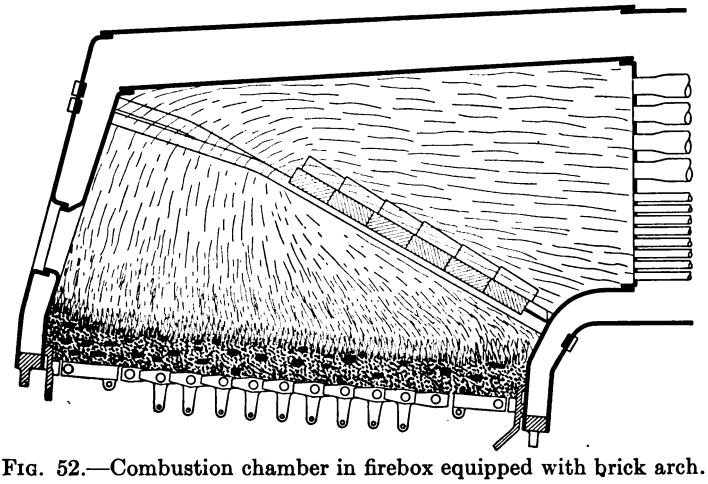


FIG. 52.—Combustion chamber in firebox equipped with brick arch.

The steam engine shown in the upper left-hand corner of Fig. 51 may be run at speeds from 400 to 600 r.p.m. The claims made for this stoker are, first, its non-interference with the fire door; second, the simplicity of its construction; third, the ease with which it is operated; fourth, the control it gives the fire. It requires only a few seconds to place the stoker in operation, and, after it has been adjusted, it needs no attention of the fireman other than the operation of the mechanism.

The results of tests on mechanical stokers seem to indicate that the even distribution in thin layers of coal prevents the formation of clinkers. Under ordinary conditions, the cleaning of the fire is not found necessary.

Brick Arch.—Among the devices for the betterment of combustion in locomotives, is the brick arch, the general location of which in the fire-box is shown in Fig. 52. A later development is the hollow brick arch which allows air to be admitted above the fire to unite with the combustible gases. It was designed to supply additional air after being heated by passing through the hot passage within the hollow brick, as shown in Fig. 53.

From a series of runs made on the Altoona locomotive testing plant, applying arches of different lengths, the following important conclusions were reached:

(*A*) The use of the brick arch, with a high volatile coal, such as Penn Gas, results in an increased evaporation, representing

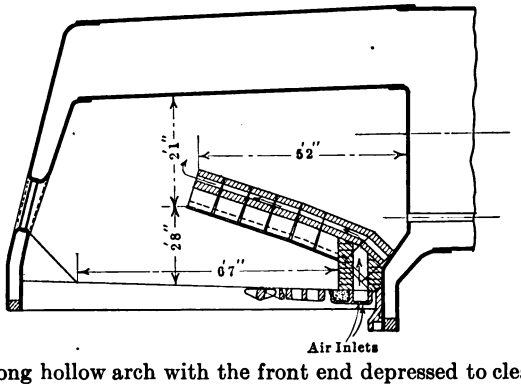


Fig. 53.—Long hollow arch with the front end depressed to clear the tubes. This arch was used with the air inlets both closed and open and with part of the rear portion of the grate covered with firebrick. This arch showed the best results and there was least smoke when the air passages were closed.—P. R. R. Tests.

in coal economy of from 12 to 13½ per cent., the indication being that the hollow arch has no advantage over the solid one.

(*B*) With a low volatile coal, such as Scalp Level, the arch does not appear to be of much benefit.

(*C*) The admission of air through the arch does not appear to decrease the amount of smoke as obtained with the solid arch.

(*D*) The smoke from a smoky coal, such as Penn Gas coal, can be reduced by the use of the arch so that it is less than the smoke from a low volatile coal without an arch, but it cannot be made so little as was obtained with low volatile briquettes without an arch.

(*E*) The best results were obtained with the long arch and with air admitted to the fire-box through the fire door. The in-

crease in economy and decrease in smoke followed closely the increase in the length of the arch.

Powdered Coal.—It has been pointed out earlier in this chapter that to prevent smoke and secure better fuel economy the fuel should be continuously supplied in small quantities and the gases distilled from the coal at a uniform rate. One cubic foot of solid coal exposes about 6 sq. in. for absorption and liberation of heat, while a cubic inch of powdered coal exposes from 20 to 25 sq. ft. The possibilities in the use of pulverized fuel are many, notwithstanding the difficulties to be overcome. The cost of pulverizing and drying the fuel is in part offset by the possibility of using a lower grade of fuel. Mr. W. L. Robinson, Supervisor of Fuel Consumption, Baltimore & Ohio, has noted the following considerations:[1] It enables the more uniform gas production from the volatile matter in the coal and the more prompt and perfect intermingling of gas and air, thereby improving combustion and reducing smoke. Furthermore, there is no cooling of the fire by heavy intermittent charges of fresh coal, and consequent production of lost heat.

The mechanical production of coal by machine undercutting and shearing and by powder mining has materially affected the grade of coal produced, by increasing the percentage of dust and slack, which in some instances is as high as from 45 to 55 per cent. of the total mine output. The annually increasing expense to produce the inferior qualities and grades of coal in the mining operations, now make it essential that the railways utilize as much as possible of the inferior grades and qualities of local fuel supply available, in order to conserve the better mine output for commercial revenue tonnage in the domestic and foreign trade.

It is claimed that to give the best results for complete combustion and the least trouble as regards ash and slag, powdered coal should contain not more than 1 per cent. moisture, and be of a uniform fineness, so that not less than 95 per cent. will pass through a 100-mesh, and not less than 85 per cent. through a 200-mesh, and not less than 70 per cent. through a 300-mesh screen. The cost for preparing powdered coal will vary with the cost for the raw coal and its moisture content. However, a general average from available data covering periods of the past 5 to 10 years at cement and metallurgical plants will enable the following conservative estimate, assuming the cost of the

[1] Railway Age Gazette, May 21, 1915.

raw coal at from $1 to $2 per short ton, and that it will require crushing and have a moisture content of from 5 to 10 per cent. when placed in the dryer.

Capacity of plant in short tons per hour	Average total cost for preparation per short ton
2	From 25 to 50 cents
3	From 20 to 45 cents
4	From 16 to 40 cents
5	From 14 to 35 cents
10	From 12 to 30 cents
25	From 10 to 20 cents

The fuel required for drying the coal will average from 1 to 2 per cent. of the coal dried. The distribution of the total cost may be approximately stated as:

Fuel for drying	10 per cent.
Power for operation	30 per cent.
Labor	30 per cent.
Maintenance and supplies	25 per cent.
Interest, taxes, insurance and depreciation	5 per cent.
Total	100 per cent.

Finely divided coal dust gives off gas at normal atmospheric temperature, but any pulverized coal coarser than that which will pass through a number 100-mesh screen is liable to explosion only when distilled by the heat or compression of a primary ignition.

Powdered coal may be burned by either of two generally defined methods: The first, or long-flame method, constitutes a progressive burning of the coal. This combustion is accomplished by projecting the primary air which carries the fuel into the furnace with high velocity, the additional air (about 75 per cent.) required for combustion being blown or induced into the furnace from other sources. The second, or short-flame method, has been the latest development. This process involves a flame of relatively short travel, and consists of admitting the entire air supply needed for combustion into the furnace with the fuel at low velocity. In the application of powdered coal to a New York Central locomotive, a combination of the long and short-flame methods has been used.

The principal requirements are: An enclosed fuel container; means for conveying the fuel to the feeders; means for commingling the fuel with air at the time of and after feeding; supplying the proper amount of air to produce a combustible mixture at

the time the fuel and air finally enter the furnace; a suitable refractory-material furnace in the fire-box; means for disposing of the slag; means for producing the proper draft through the furnace and the boiler; means for harmonizing the draft and the combustion; suitable power for operating the fuel and air feeding mechanism, and an automatic and hand control of the fuel and air regulation.

The use of bituminous coal in a powdered form seems to be the logical solution of the smoke, cinder and spark questions at engine houses, terminals and on the road, and one that should greatly reduce the loss of heat and fuel cost resulting from imperfect combustion in existing and future steam locomotives. The elimination of ash pans, grates, smoke-box, diaphragm, baffles and nettings substantially reduces the retardation of the products of combustion through the boiler.

With powdered coal, the fuel is supplied to an enclosed, air-tight container on the tender (suitable for either powdered or liquid fuel), prepared to uniform fineness and thoroughly dried, so that when fed to the furnace it immediately produces effective heat. Furthermore, the coal is not touched by hand or shovel from mine car to furnace and there is no loss by pilfering, dropping from the tender container, gangways, through holes in deck, or by firemen shoveling undesirable fuel off the tender on right-of-way. When powdered fuel is used, the refractory-material furnace retains its heat and prevents the chilling of the fire-box and flues, even though the supply of fuel may be cut off and therefore reduces the liability of fire-box leakage.

When being worked at from one-half to maximum boiler horsepower capacity, a locomotive boiler equipped with a superheater will range from 65 to 55 per cent. boiler efficiency, this being representative of the best grate fire practice. Taking into consideration the effect of burning powdered coal in suspension, on the various heat losses enumerated, it is conservative to place the saving to be effected at 25 per cent. of the coal fired, actual performance to date having shown as high as from 30 to 40 per cent. saving.

Design of Locomotives with Respect to Fuel Economy.[1]—That design of a locomotive boiler and fire-box, together with the appurtenances, which will permit of the largest possible amount of evap-

[1] For further details see Report of Committee on "Fuel Economy," American Railway Master Mechanics Asso., June, 1915.

oration from a given amount of combustible burned, has the maximum efficiency, and is, therefore, the best boiler from the standpoint of fuel economy. This statement defines an ideal for the attainment of which is being approached more nearly as time goes on.

Unless a locomotive has been designed according to the best-known practices, it cannot be expected to show as great economies in the use of fuel as would a locomotive which had been properly proportioned. Greater attention is being given to the design of boilers, fire-boxes, grates, ash pans and front ends than ever before, as all of these parts are interdependent and should bear a definite relation for the best results, and are influenced by the nature of the fuel available.

A study of certain ratios in locomotives constructed within the past 2 years, and in others built 10 or 12 years ago, shows the trend of locomotive design in those respects which have the greatest influence on fuel consumption and steaming capacity. The data shown below are the averages for a large number of bituminous and anthracite burning locomotives of the 4–6–2, 4–6–0, 2–8–0 and 0–6–0 types.

1.—BITUMINOUS

Type of locomotive	Ratio total evaporative heating surface to grate area		Ratio fire-box heating surface to total evaporative heating surface	
	1913	1903	1913	1903
4–6–2	65.9	70.9	6.4	5.4
4–6–0	52.3	62.8	7.8	6.5
2–8–0	53.3	60.1	6.7	5.7
0–6–0	53.0	71.8	8.3	6.0
Average	56.1	66.4	7.3	5.9
Per cent. change	Dec. 15.5	per cent.	Inc. 23.7	per cent.

2.—ANTHRACITE

4–6–0	30.0	34.8	8.0	6.5
2–8–0	31.1	37.9	8.3	6.1
Average	30.5	36.3	8.1	6.3
Per cent. change	Dec. 16 per	cent.	Inc. 28.5	per cent.

The changes in these ratios are, without doubt, steps in the right direction. The reduction in the ratio of total evaporative heating surface to grate area means that to produce a given

amount of evaporation, less coal will have to be burned per square foot of grate area, resulting in a higher fuel efficiency, and an increased fuel capacity when it becomes necessary to force the boiler. A distinct gain in fuel efficiency is represented by the increase in the ratio of fire-box heating surface to total evaporative heating surface, because reliable tests have demonstrated that a square foot of fire-box heating surface will evaporate about five times as much water per hour as will a square foot of tube heating surface, when the locomotive is developing its rated working power. This shows the importance of a relatively large fire-box heating surface in relation to fuel economy. The increase of nearly 24 per cent. for bituminous and over 28 per cent. for anthracite burning locomotives, in the ratio of fire-box heating surface to total evaporative heating surface, is a marked improvement in this direction.

The advantage of large heating surfaces and large grate area in increasing the evaporation from a unit amount of fuel consumption over that from a similar locomotive with smaller heating surfaces and grate area, is clearly shown by an evaporation test recently conducted at the Pennsylvania Railroad testing plant at Altoona, between a Consolidation type and a Mikado type locomotive. The principal dimensions of the two locomotives and a summary of the data follow:

TABLE X.—PRINCIPAL DIMENSIONS

Item	Consolidation	Mikado	Per cent. increase favor of Mikado
Cylinders	25 by 28 in.	27 by 30 in.	25 in volume
Drivers	62 in.	62 in.	.....
Boiler pressure	205 lb.	205 lb.	.....
Smallest diameter of boiler.	76¾ in.	76⅝ in.	.....
Length of flues and tubes..	15 ft. 0 in.	19 ft. 0 in.	25.7
Heating surface, flues and tubes.	2,537.7 sq. ft.	3,374.6 sq. ft.	.....
Heating surface, superheater.	809.0 sq. ft.	1,171.6 sq. ft.	44.8
Heating surface, fire-box..	189.9 sq. ft.	301.5 sq. ft.	58.8
Heating surface, total.....	3,536.6 sq. ft.	4,847.7 sq. ft.	37.1
Grate area	55.1 sq. ft.	70.0 sq. ft.	26.5
Weight on drivers	219,900 lb.	235,800 lb.	7.2
Total weight of engine....	249,599 lb.	315,600 lb.	26.5
Total tractive power.....	46,290 lb.	57,850 lb.	25.0

SUMMARY OF TEST DATA

Dry coal fired per hour, pounds	Water evaporated, per hour, pounds		Per cent. increase favor of Mikado
	Consolidation	Mikado	
2,000	17,000	17,000	0.0
4,000	27,000	30,000	11.1
6,000	32,500	43,000	32.3
8,000	34,500	52,500	52.2

These figures clearly show the advantage of the large heating surface and large grate area, and that the economy increases with a decrease in the rate of combustion.

The above testing-plant figures are verified by the results obtained from a road test on the Chicago & North Western, reported in the Railway Age Gazette of January 15, 1915, pages 93–94.

Operation of Superheater Locomotives.—When properly maintained and efficiently operated, the superheater is by far the most valuable mechanical aid to fuel economy ever applied to locomotives. By its use savings of 20 to 25 per cent. in coal and water are obtainable in actual service. But if maintenance is neglected and careless handling of the device in service is permitted, the superheater may become almost useless, as far as performing its regular function is concerned.

Actual test data prove conclusively that greater economy of fuel and water is obtained by operating with a full throttle opening and a short cut-off than with a partial throttle opening and a longer cut-off. The following table is from the Report of Tests of a Class E6s passenger locomotive at Altoona.

TABLE XI

	Lb. per indicated horse power hour at 1500 hp.			Per cent. saving	
	Partial throttle		Full throttle		
	1 40 per cent. cut-off	2 30 per cent. cut-off	3 20 per cent. cut-off	3 over 1	3 over 2
Dry coal, pounds.	2.52	2.25	2.02	19.8	10.2
Steam, pounds...	18.9	17.5	15.8	16.4	9.7

TEMPERATURE OF GASES IN BOILER TUBES

During a series of tests in 1912 and 1913, on the locomotive testing plant at Altoona, observations were made of the temperature in saturated steam tubes and in superheater flues, average results of which are shown in Fig. 54.[1] In establishing a general equation to show the law of change of temperature, the author has followed a process suggested by Dr. E. R. Smith, of the Department of Mathematics, Pennsylvania State College. The ultimate aim of determining the drop in temperature is

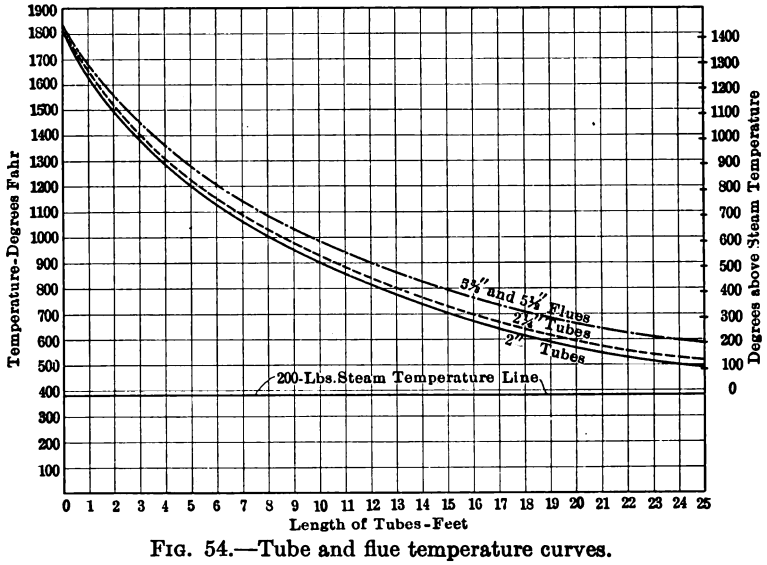


FIG. 54.—Tube and flue temperature curves.

Average temperature readings are here plotted, which show the heat of gases in 2-in. and 2¼-in. (outside diameter) tubes, 5¾-in. and 5½-in. flues, in passing from firebox to smokebox. Rate of combustion per square foot of grate per hour about 100 to 115 lb.

to determine the most economical length of tube in a particular type of locomotive. An important series of results is shown in Fig. 55. In the locomotive tested, the increased rate of combustion causes a material rise in temperature only in the fire-box.

The following notation is here used:

t_x = temp. of gases at a dist. x from fire-box end of tube.

t_w = temp. of water in boiler. (It is here assumed that the pressure is 200 lb. and that $t_w = 389°$, the temp. of saturated steam at that pressure.)

[1] Summarized by Mr. F. J. Cole in *Bulletin* 1017, American Locomotive Company.

t_f = temp. of gases at fire-box end of tube, *i.e.*, for $x = 0$.
t_s = temp. of gases at smoke-box end of tube (or at the point of the last observation).
x = dist. from fire-box end of tube expressed in feet.
k = constant, depending for its value on the rate of combustion, specific heat of gases, draft, etc.
l = length of flue or tube in feet.

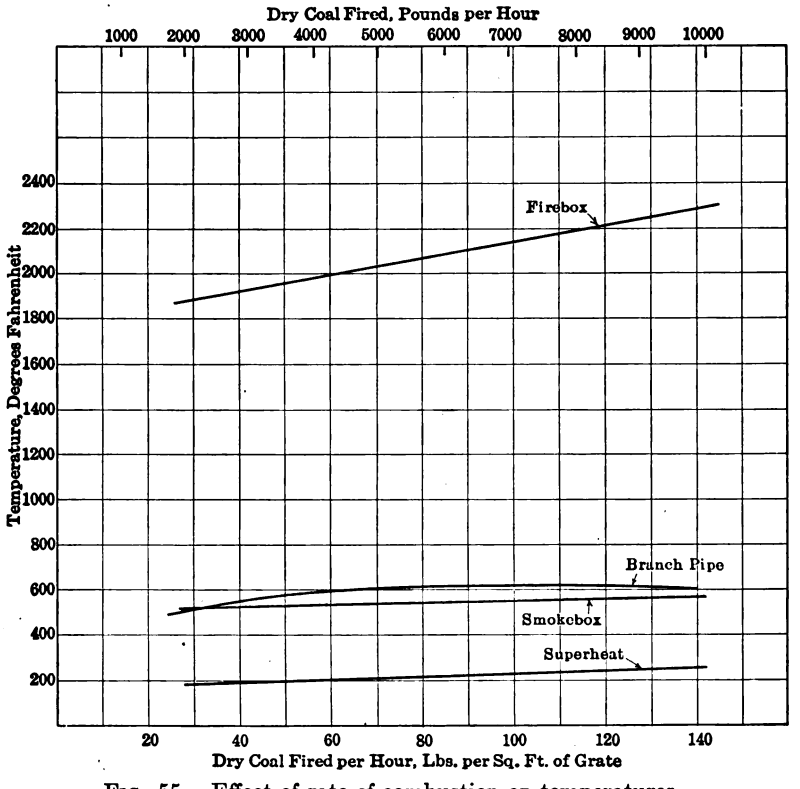


FIG. 55.—Effect of rate of combustion on temperatures.

The fire-box temperature is materially increased by the increased amount of dry coal burned. The other temperatures remain nearly constant. (From tests on K29 locomotive, Altoona test plant, 1912.)

It has been experimentally verified that the rate of decrease in temperature of the gases is proportional to the difference in the temperature of gases and the temperature of the water. That is,

$$\frac{dt_x}{dx} = k(t_x - t_w) \text{ or } \frac{d(t_x - t_w)}{dx} = k(t_x - t_w)$$

Integrating, we obtain,

$$t_x - t_w = Ce^{kx} (e = 2.7182)$$

and since for $x = 0$ we have $t_x = t_f$,

$$C = t_f - t_w$$

so that

$$t_x = t_w + (t_f - t_w)e^{kx} \tag{78}$$

The value of k can be determined from the data by the methods of least squares. If t'_x is the observed temperature for any given value of x, then k must have such a value that

$$\Sigma[\log (t'_x - t_w) - \log (t_x - t_w)]^2$$

shall be a minimum. Substituting in this expression the right-hand member of Eq. 78, we obtain,

$$\Sigma[\log (t'_x - t_w) - \log (t_f - t_w) - kx \log_{10} e)]^2$$

Differentiating with respect to k we obtain the condition which k must satisfy in the form

$$\Sigma[\log (t'_x - t_w) - \log (t_f - t_w) - kx \log_{10} e]x = 0$$

or

$$\Sigma x \log (t'_x - t_w) - \log (t_f - t_w)\Sigma x - k \log_{10} e \Sigma x^2 = 0$$

from which it follows that

$$k = \frac{\Sigma x \log (t'_x - t_w) - \log (t_f - t_w)\Sigma x}{0.4343\Sigma x^2} \tag{79}$$

A good approximation of k is given in terms of the temperatures t_f and t_w. For $x = l$, the length of the tube, we have

$$t_s - t_w = (t_f - t_w)e^{kl}$$

from which it follows that

$$k = \frac{1}{0.4343l} \log \frac{t_s - t_w}{t_f - t_w} \tag{80}$$

By referring to Table XII we see that in this case the result is very close to the value obtained by the method of least squares.

By using (78) and (80), note that the temperature of the gases at any point of the tube is determined when the pressure and the temperatures at the ends of the tubes are known.

The formula gives the temperature t_x about as accurately as do the readings of the pyrometers.

TABLE XII.—CALCULATIONS FOR 2¼" TUBE

x (in ft.)	t'_x (Fig. 54)	$\Sigma x \log (t'_x - 389)$	
0	1810	0	
1	1660	3.105	
2	1520	6.108	
3	1410	9.030	Substituting in Eq. 79,
4	1310	11.860	$k = \frac{688.944 - 3.154 \times 276}{0.4343 \times 4344}$.
5	1220	14.600	
6	1160	17.328	
7	1090	19.922	$k = 0.0962$.
8	1030	22.464	
9	980	24.948	
10	930	27.329	
11	885	29.690	Computed by approx. formula, Eq. 80,
12	840	31.800	$k = 0.1023$ (which value should be
13	800	33.900	checked by the student).
14	770	36.180	
15	730	37.980	
16	700	39.850	
17	670	41.700	
18	640	43.250	From Eq. 78,
19	620	45.000	$t_x = 389 + (1810 - 389)e^{-0.0962x}$
20	595	46.400	
21	570	47.400	Which expresses the law of tempera-
22	555	48.900	ture change in the 2¼-in. tube.
23	540	50.200	
$\Sigma x = 276$			
$\Sigma x^2 = 4344$			
$\log (t_f - t_w) = 3.154$		688.944	

Problem 23.—Using values for *flues* from Fig. 54, determine t_x in Eq. 78.

QUESTIONS ON CHAPTER VIII

1. What are the advantages and disadvantages of air excess? Does excess air raise or lower the carbon monoxide?

2. Name five causes which may account for dense black smoke in a locomotive.

3. What determines the proper grate area to use?

4. From a study of Table VIII, what would be the most desirable way to increase the over-all efficiency?

5. On the basis of the theory outlined in this chapter, give a summary of the chief advantages of (*a*) the overfeed and (*b*) the underfeed mechanical stoker.

6. What are the chief considerations in the design of locomotives for the best fuel economy?

7. Referring to Fig. 54, name two reasons why the rate of temperature drop is more rapid during the first 10 ft. than it is during the second 10 ft. of length from fire-box end.

Problem 24.—Special review problems, involving change of speed, braking, energy from coal, etc. (slide-rule calculations).

A train consisting of locomotive, tender and twelve *P*-70 loaded passenger cars is traveling on a uniform 2 per cent. up-grade at 30 m.p.h. Brakes are applied with steam shut off and the speed reduced to 20 m.p.h. With full boiler steam capacity, it is again accelerated to 30 m.p.h.

(*a*) How many minutes were lost, caused by this slow-down? Assume uniform acceleration and retardation and solve from the following data for a Pacific type locomotive.

(*b*) How many pounds (extra) of coal required in regaining the speed?

Data: 21″ × 26″ cylinders, 63″ drivers	200 lb. boiler press.
Total heating surface	3,025 sq. ft.
Weight of locomotive in running order	167,000 lb.
Weight of tender, loaded	110,000 lb.
Weight of locomotive on drivers	130,000 lb.
Capacity of tender, coal	10 tons.
Capacity of tender, water	5,400 gallons
Weight of cars, empty, each	61.2 tons
Weight of cars, loaded each	65 tons

Light-weight tender = $110{,}000 - (5400 \times 8\tfrac{1}{3} + 10 \times 2000) = 22.5$ tons

Solution: (*a*).—

To stop train:

Light weight	Per cent. braked	Friction	Loaded weights
Loco. 130,000	× 0.60 × 0.10 =	7,800	167,000 lb.
Tender 45,000	× 0.80 × 0.10 =	3,600	110,000 lb.
12 cars (lt. wt.) 1,400,000	× 0.90 × 0.10 =	126,000	1,560,000
Total		137,400	1,837,000

= 918.5 tons

NOTE.—Average coefficient of friction, taken as 0.10, is a conservative value.

$$\frac{137{,}400}{1{,}837{,}000} = 0.0742 = 7.42 \text{ per cent.}$$

$\frac{1}{6}$ per cent. up-grade = 0.20 lb. resistance per ton which is to be *added* to 7.42 = 7.62. From Eq. 54, Chap. VIII,

$$S = 3.34 \times \frac{V_2^2 - V_1^2}{F\% + G} = 3.34 \frac{30^2 - 20^2}{7.62} = 219 \text{ ft.}$$

This may be checked by Eq. 19, Chap. IV, remembering to add the effect of grade.

$$S = 66.8 \times \frac{900 - 400}{\dfrac{137{,}400 + 0.4 \times 918.5}{918.5}} = 222 \text{ ft.}$$

From Eq. 52, $t = 4.78 \times \frac{30 - 20}{7.62} = 6.28$ sec. consumed in stopping.

Tractive force (from Eq. 11, Chap. III) for average velocity = $\frac{0.85Pd^2s}{D} \times$ speed factor.

For 25 m.p.h., from Fig. 11, speed factor = 0.60.

T.F. = 30,900 × 0.60 = 18,540

Resistance, engine and tender	= 138.5 × 6.13 + (20 × 0.2)	= 1427 lb
Resistance, twelve cars	= 780 × (5.2 + 4)	7167 lb.
		8594 lb.

Available T.F. = 18,540 − 8594 = 9946 lb.

$$\frac{1}{a} = \frac{W}{Fg} = \frac{2000 \times (138.5 + 780) \times 1.05}{9946 \times 32.16} = 6.00$$

The value 1.05 is the 5 per cent. added to the weight, equivalent to the retarding effect of inertia of the wheels.

(The value of $\frac{1}{a} = 6$ corresponds to a point at 25 m.p.h. on the reciprocal curve used in the speed-time analysis of train operation. The solution of this problem could be worked from here by graphical method if desired.)

Proceeding:

$$t = \frac{1}{a} \times (v_2 - v_1)$$

$= 6.00 \times (44 - 29.25) = 88.5$ sec. to accelerate from 20 to 30 m.p.h.

88.5 + 6.28 = 94.78 sec. to slow down and to accelerate.

$$S = vt = \frac{44 + 29.25}{2} \times 88.5 = 3240 \text{ ft.}$$

3240 + 222 = 3462 ft.

$\frac{3462}{44}$ = 79 sec. to cover 3462 ft. at 30 m.p.h. had this speed been uniformly maintained. 94.78 − 79 = 15.78 sec. lost. *Ans.*

Solution: (*b*).—The kinetic energy K (or the work done) required to accelerate the mass of 1,837,000 lb. from a velocity V of 29.25 ft. per sec. (30 m.p.h.) to $v_2 = 44$ ft. per sec. (40 m.p.h.) is equal to $K = \frac{W(V_2^2 - V_1^2)}{2 \times 32.16} = \frac{1{,}837{,}000(1936 - 856)}{64.32} = 31{,}000{,}000$ ft.-lb.

Assume 1 lb. of coal contains 13,000 B.t.u. per lb. Since 778 ft.-lb. equal 1 B.t.u., 13,000 × 778 = 10,114,000 ft.-lb. available in 1 lb. coal. Of this, about $\frac{1}{20}$ appears as useful work at the track, or

$$\frac{10{,}114{,}000}{20} = 505{,}700 \text{ ft.-lb.}$$

31,000,000 ÷ 505,700 = 61.5 lb. *Ans.*

The solution of this problem, as here given, brings out many of the principles outlined in former chapters, but does not go into refinements of method or analysis, which is not usually required unless more conditions than here assumed are accurately known.

Problem 25.—Using the data for a K2 locomotive, Table II, hauling thirty 100,000 lb. capacity, fully loaded cars (light weight 45,000 lb.) running at 40 m.p.h. down a 0.20 per cent. grade, find the time lost if brakes are suddenly applied and the train retarded until the speed is 30 m.p.h.; the train is then uniformly accelerated to 40 m.p.h.

Problem 26.—If the train in Prob. 24, including locomotive and tender, coasted down a 1 per cent. grade 1000 ft. long without steam or brakes applied, find the distance travelled before it would come to rest on a level straight track at the bottom of the grade.

CHAPTER X

STEAM: ITS FORMATION AND ACTION

RAPID DEVELOPMENT DURING THE PAST 10 YEARS; TYPES OF LOCOMOTIVE BOILERS; ELEMENTARY PRINCIPLES OF HEAT REVIEWED; PROPERTIES OF STEAM; EQUIVALENT EVAPORATION; TRANSFER RATE OF HEAT OF GASES TO STEAM; RELATIVE EVAPORATION RATES OF FIRE-BOX AND TUBES; ACTION OF STEAM IN THE CYLINDER; DISCUSSION OF CYCLES; LOSSES IN ACTUAL INDICATOR CARD; TYPICAL INDICATOR CARDS; EFFECT OF PISTON SPEED AND HIGH PRESSURE; BACK-PRESSURE; PROBLEM IN DESIGN OF BOILER; VALVE MOTION; WALSCHAERTS VALVE GEAR

Introductory.—The beginning of the nineteenth century found the steam engine fully developed in all of its principal features. When, in 1804, Oliver Evans, the "Watt of America" and Richard Trevithick adapted the steam engine to the propulsion of carriages and, in 1815, George Stephenson developed his first locomotive, the following properties of steam were known: (1) water, in changing to steam, consumes a definite quantity of heat; (2) saturated steam exists in that state at a definitely fixed temperature for each pressure; (3) the volume of a pound of steam varies with the pressure; (4) steam may be used expansively. In the early days, steam economy of locomotives was given but little consideration; in fact, only within a few years has the saving in water—thus indicating a saving in coal—become a matter of moment.

Since the beginning of the twentieth century the Walschaerts valve gear, high steam pressure and superheated steam have become the fixed practice in this country.

The formation and the action of steam are the most vital features of the locomotive. The first chapter traced briefly the journey of steam from the steam dome to the cylinders, thence to the atmosphere. Following the principles outlined for good combustion, in Chap. IX, we are now concerned with the economical use of the heat conducted to the water and steam in contact with the hot metal surfaces.

TABLE XIII.—PROPERTIES OF SATURATED STEAM

(From Marks and Davis' Tables)

Pressure above atmosphere		Temperature		British thermal units above 32°F.			Volume of 1 lb. in cubic feet
Absolute	Gage	Fahrenheit	Centigrade	Heat of the water	Latent heat of steam	Total heat of steam	
P	*G*	*F*	*C*	*h*	*L*	*H*	*V*
14.7	0	212.0	100.00	180.0	970.4	1150.4	26.79
16.7	2	218.5	103.61	186.6	966.2	1152.8	23.80
18.7	4	224.4	106.78	192.5	962.4	1154.9	21.36
20.7	6	229.8	109.88	198.0	958.8	1156.8	19.45
22.7	8	234.8	112.66	203.0	955.6	1158.6	17.85
24.7	10	239.4	115.22	207.7	952.5	1160.2	16.49
26.7	12	243.7	117.61	212.1	949.6	1161.7	15.34
28.7	14	247.8	119.89	216.2	946.8	1163.0	14.33
30.7	16	251.6	122.00	220.1	944.2	1164.3	13.45
32.7	18	255.3	124.05	223.8	941.7	1165.5	12.68
34.7	20	258.8	126.00	227.4	939.3	1166.7	11.99
36.7	22	262.1	127.83	230.8	936.9	1167.7	11.38
38.7	24	265.3	129.61	234.0	934.7	1168.7	10.82
40.7	26	268.3	131.28	237.2	932.5	1169.7	10.32
42.7	28	271.3	132.94	240.1	930.5	1170.6	9.87
44.7	30	274.1	134.50	243.0	928.5	1171.5	9.45
46.7	32	276.8	136.00	245.7	926.6	1172.3	9.07
48.7	34	279.4	137.44	248.4	924.7	1173.1	8.72
50.7	36	281.9	138.83	251.0	922.9	1173.9	8.40
52.7	38	284.3	140.17	253.5	921.1	1174.6	8.10
54.7	40	286.7	141.50	256.0	919.3	1175.3	7.82
56.7	42	289.0	142.77	258.3	917.6	1175.9	7.56
58.7	44	291.3	144.05	260.6	915.9	1176.6	7.32
60.7	46	293.5	145.28	262.9	914.3	1177.2	7.09
62.7	48	295.6	146.44	265.1	912.7	1177.8	6.88
64.7	50	297.7	147.61	267.2	911.2	1178.4	6.68
74.7	60	307.3	152.94	277.1	903.9	1181.0	5.83
84.7	70	316.0	157.78	286.1	897.3	1183.3	5.18
94.7	80	323.9	162.17	294.3	891.1	1185.3	4.66
104.7	90	331.2	166.22	301.8	885.4	1187.1	4.241
114.7	100	337.9	169.94	308.8	880.0	1188.8	3.890
120.7	106	341.7	172.05	312.8	876.9	1189.7	3.706
126.7	112	345.4	174.10	316.6	873.9	1190.5	3.538
132.7	118	348.9	176.05	320.3	871.0	1191.3	3.385
138.7	124	352.4	178.00	323.8	868.2	1192.1	3.248
144.7	130	355.7	179.83	327.3	865.5	1192.8	3.118
150.7	136	358.9	181.61	330.6	862.9	1193.5	2.999
156.7	142	362.0	183.33	333.9	860.3	1194.2	2.890
162.7	148	365.0	185.00	337.1	857.7	1194.8	2.790
168.7	154	367.9	186.61	340.1	855.2	1195.4	2.695
174.7	160	370.7	188.18	343.1	852.8	1195.9	2.606
180.7	166	373.5	189.72	346.0	850.5	1196.5	2.524
186.7	172	376.2	191.22	348.9	848.1	1197.0	2.447
192.7	178	378.8	192.67	351.6	845.9	1197.5	2.373
195.7	181	380.1	193.40	353.0	844.8	1197.8	2.338
198.7	184	381.3	194.05	354.3	843.7	1198.0	2.304
201.7	187	382.6	194.78	355.7	842.5	1198.2	2.272
204.7	190	383.9	195.50	357.0	841.5	1198.5	2.240
207.7	193	385.1	196.17	358.3	840.4	1198.7	2.210
210.7	196	386.3	196.83	359.5	839.4	1198.9	2.180
214.7	200	387.9	197.72	361.3	838.0	1199.2	2.141
219.7	205	389.8	198.7	363.3	836.1	1199.6	2.094
224.7	210	391.8	199.89	365.4	834.5	1199.9	2.049
234.7	220	395.5	201.94	369.3	831.2	1200.6	1.966
244.7	230	399.2	204.00	373.2	828.0	1201.2	1.889
254.7	240	402.6	205.88	377.0	824.8	1201.7	1.817
264.7	250	406.1	207.84	380.6	821.7	1202.3	1.751
274.7	260	409.4	209.67	384.1	818.7	1202.8	1.689

TABLE XIV.—PROPERTIES OF SUPERHEATED STEAM
(From Tables by Marks and Davis)

Boiler pressure, lb. per sq. in., *P*	Superheat, degrees F.	Temp. of steam, degrees F., *F*	Total heat B.t.us. per pound *H*	Specific volume, cubic feet per pound, *V*	Increase in volume over sat. steam, per cent.
150	50	415.9	1225.2	2.99	8.7
	75	440.9	1238.8	3.10	12.7
	100	465.9	1252.0	3.21	16.7
	150	515.9	1277.6	3.43	24.7
	200	565.9	1302.5	3.64	32.4
160	50	420.7	1226.6	2.83	8.9
	75	445.7	1240.3	2.93	12.6
	100	470.7	1253.6	3.04	16.9
	150	520.7	1279.1	3.24	24.6
	200	570.7	1304.1	3.44	32.3
170	50	425.3	1227.9	2.68	8.5
	75	450.3	1241.7	2.78	12.5
	100	475.3	1255.0	2.89	17.0
	150	525.3	1280.6	3.08	24.8
	200	575.3	1305.6	3.27	32.4
180	50	429.6	1229.2	2.55	8.5
	75	454.6	1243.1	2.65	12.8
	100	479.6	1256.4	2.75	17.0
	150	529.6	1282.0	2.93	24.7
	200	579.6	1307.0	3.11	32.3
190	50	433.8	1230.4	2.44	8.9
	75	458.8	1244.4	2.53	13.0
	100	483.8	1257.7	2.62	17.0
	150	533.8	1283.3	2.80	25.0
	200	583.8	1308.3	2.97	32.7
200	100	487.9	1259.0	2.51	17.3
	150	537.9	1284.6	2.68	25.2
	200	587.9	1309.7	2.84	32.7
	250	638.0	1334.0	3.00	38.1
	300	688.0	1359.1	3.16	46.9
205	100	489.9	1259.6	2.45	17.2
	150	539.9	1285.2	2.62	25.3
	200	589.9	1310.3	2.78	33.0
	250	639.9	1335.1	2.94	40.7
	300	689.9	1359.8	3.10	48.3

In Fig. 56 is shown a Belpaire type of boiler, with names of the parts. The student should study carefully this sketch, noting especially the relative location of the boiler tubes, super-

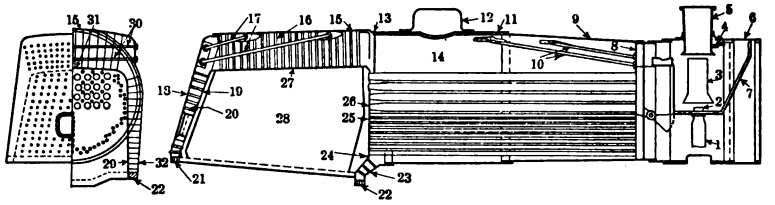


FIG. 56.—Fire-tube boiler, Belpaire type.

Parts of a locomotive boiler

1. Exhaust pipe.
2. Exhaust nozzle.
3. Lift or Petticoat pipe.
4. Stack extension.
5. Smoke stack.
6. Smoke box.
7. Spark arresting netting.
8. Front tube sheet.
9. First course.
10. Braces, front tube sheet.
11. Second course or dome sheet.
12. Dome.
13. Hip sheet.
14. Cylindrical part or barrel.
15. Roof sheet.
16. Crown bolts.
17. Braces, back head.
18. Back head.
19. Door sheet.
20. Stay bolts.
21. Mud ring.
22. Mud ring.
23. Outside throat sheet.
24. Back tube sheet.
25. Fire tubes.
26. Superheater flues.
27. Crown sheet.
28. Fire-box.
29. Inside side sheet.
30. Cross stays.
31. Crown sheet.
32. Outside side sheet.

heater flues, exhaust pipe, steam dome and mud ring. Observe the method of staying the crown sheet and of bracing the front tube sheet and the back-head. The nearly flat and parallel

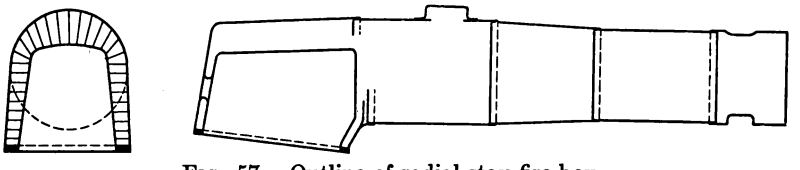


FIG. 57.—Outline of radial stay fire-box.

inside and outside sheets allow straight and direct staying in this type. This boiler is used commonly on the 2–8–0 and 2–8–2 locomotives.

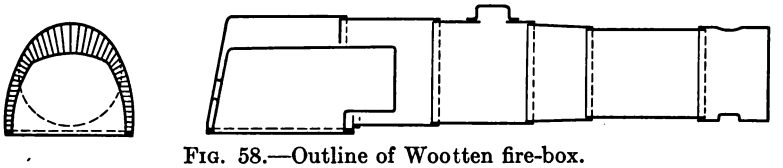


FIG. 58.—Outline of Wootten fire-box.

The above is but one of a number of types of locomotive boilers. The radial-stay boiler derives its name from the construction of the crown sheet stays and bolts, which are set on lines which form

radii of the inner and outer sheets, as shown in the sketch, Fig. 57. The Wootten fire-box, Fig. 58, has a curved crown and roof sheet of large radius.

The wide fire-box boiler is a general term applied to a fire-box resting on the frame and extending out over the trailing wheels, thus providing for a wide grate. This notable development in design enables the steaming capacity of locomotives to be increased to meet the most exacting demands of locomotives in high-speed passenger and freight service, and thus makes it unnecessary to force the combustion up to the very high rates, as 200, or even more pounds per square foot per hour. A large boiler not only gives an ample supply of steam but secures that supply

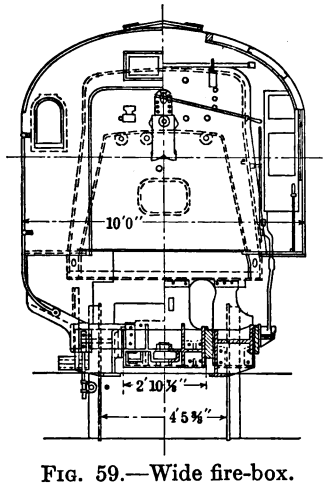


FIG. 59.—Wide fire-box.

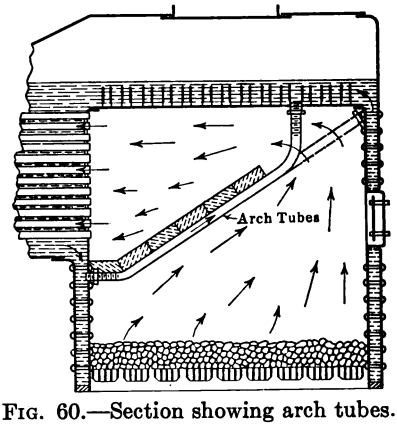


FIG. 60.—Section showing arch tubes.

with the minimum weight of fuel. The end elevation of a wide fire-box on a Pacific type locomotive is shown in Fig. 59.

When a brick arch is used, arch tubes are applied to support the fire-brick as shown in Fig. 60, and water in the boiler circulates through the tubes which thus become part of the heating surface. They are fastened to the front and back ends of the fire-box or into the crown sheet, the former construction being preferred. The water circulates in the tube as indicated by the arrows.

One of the new departures from established design is found in the Jacobs-Schupert fire-box shown in Figs. 61 and 61*a*. The usual side sheets, crown sheet, outside and roof sheets are replaced by a series of semi-elliptic sections riveted together with their flanges

away from the fire. Figure 61*a* shows three sections of the boiler with the inner channel sections in place and ready for the outer sheets. The vertical stay sheets are partially cut out and so perforated that a free circulation of water is possible throughout the boiler. This construction eliminates the necessity for stay bolts. Under the most severe low-water tests, made in 1912, a boiler of this design did not fly to pieces, even when the water was all evaporated by means of a forced fire in the fire-box.

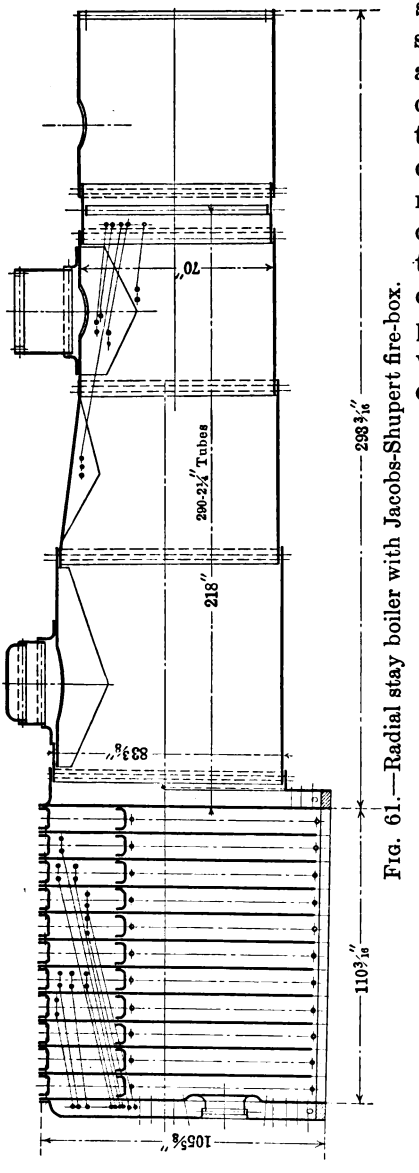
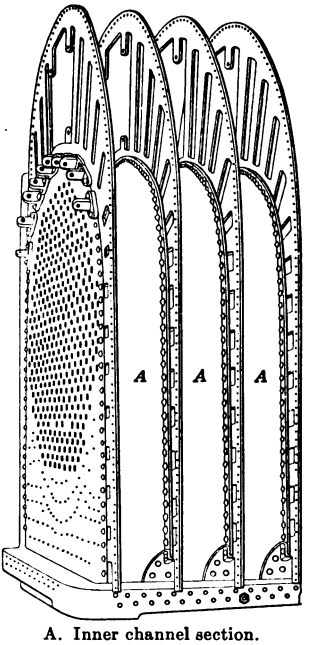


FIG. 61.—Radial stay boiler with Jacobs-Shupert fire-box.



A. Inner channel section.

FIG. 61*a*.—Section from the Jacobs-Shupert fire-box, ready for application of the outside sections.

Definitions.—In this and in the following chapter, a few definitions are given, although the principles are found in text and

reference books. In the following, the *unit* is a pound of the substance.

The *Heat of the Liquid* is the heat required to raise the temperature of water from 32°F. to the temperature at boiling point (h, in Table XIII).

The *Latent Heat* is the heat required to change water at the temperature of the boiling point into steam at that same temperature (L, in Table XIII).

Dry Saturated Steam is steam without moisture at the temperature of the boiling point corresponding to the pressure.

Wet Steam is saturated steam containing moisture.

The *Total Heat of Saturated Steam* is the heat required to raise the temperature of 1 lb. water from 32°F. and to change it into steam at the temperature of boiling point (H, in Table XIII, and this equals $h + L$).

The *Heat of Superheat*, as explained in the next chapter, is the heat required to change the temperature of saturated steam to some higher temperature while the pressure remains constant.

A *Boiler Horse Power* is equivalent to 34.5 lb. of water evaporated per hour from a feed-water temperature of 212° into steam at the same temperature.

The *Heating Surface* in a locomotive boiler, as generally used in calculations, is the "Total based on inside of fire-box, fire side of tubes, arch pipes and superheater, square feet." Some companies have used the water side of the tubes instead of the fire side, which gives a considerably large total heating surface; in the K2 locomotive, this increase is 492 sq. ft. For uniformity, all calculations should be based on fire side of *all* the surfaces. In comparing superheated steam locomotives, the "equivalent heating surface" is taken and this equals the total evaporative heating surface plus 1.5 times superheating surface.

From the *Saturated Steam Tables* (Table XIII) we find the following properties, to which special reference will be made.

TABLE XV

Gage Pres., G	Temp., °F.	Heat of water, h	Latent heat steam, L	Total heat, H	Vol. 1 lb., cu. ft.
100	337.9	308.8	880.0	1188.8	3.89
148	365.0	337.1	857.7	1194.8	2.79
200	387.9	361.3	838.0	1199.2	2.14
250	406.1	380.6	821.7	1202.3	1.75

Feed-water.—When water passes through the injector on its way to the boiler, its temperature is raised by the steam with which it mingles, the heat required being originally obtained from the boiler. Water entering the boiler must be further raised in temperature before it can be evaporated.

Saturated steam at 200 lb. gage pressure (214.7 lb. absolute) has a temperature of 387.9°F. If water enters the injector at 50°F.,the total heat in the liquid above 50°, or 361.3 − (50 − 32) = 343.3 B.t.u., must be added to each pound before it begins to vaporize. At that pressure, it requires 838 B.t.u. to change the 1 lb. into steam. This gives a total of 838 + 343.3 = 1181.3 B.t.u., which amount of heat must be added from feed-water temperature of 50°F.; that is, 29 per cent. of the total heat is absorbed in simply raising the temperature of the water to the point of vaporization. How much could this percentage be reduced by using a feed-water heater? If such a heater could be applied so as to use the heat otherwise wasted (partly from the steam exhaust of air pumps) and if the temperature of the water is thus raised to 160°F., the above percentage becomes 21 and the calculated fuel saving is $\frac{100\ (160 - 50)}{1181.3 + (32 - 50)} = 9.46$ per cent.

Exercise: (*a*) Verify the value of 21 per cent. as given, when a feed-water heater is used.

(*b*) Justify the use of the above expression for determining fuel economy.

Saving by Superheat.—Certain properties of steam offer other opportunities for saving. For example, suppose that the pound of steam after being completely vaporized at 200 lb. gage, and 50°F. feed, is superheated 200°F. (above 387.9°F.), 110.8 B.t.u. will be absorbed in superheating, which is an addition of heat equal to 100.8 ÷ 1181.3 = 0.094 or 9.4 per cent.; but superheat increases the volume of the pound 32.7 per cent.

Exercise: How many additional B.t.us. are required to superheat through 200°F. for 180 lb. gage, over that required for 200°F. superheat and 160 lb. absolute?

Equivalent Evaporation.—Boiler trials are based on different conditions of feed-water temperature and steam pressure, so that for purposes of comparison it is necessary to reduce all results to a common basis, termed "the equivalent evaporation from and at 212°F." Under this standard condition, steam would be generated *at* a temperature of 212° (the temperature of steam corre-

TABLE XVI.—ACTUAL EVAPORATION PER BOILER HORSE POWER AND FACTORS OF EVAPORATION

(Based on Marks and Davis' Tables)

Feed tem.	Gage pressure				
	125	150	175	200	250
40	28.28	28.21	28.16	28.12	28.05
	1.220	1.223	1.225	1.227	1.230
50	28.51	28.44	28.40	28.35	28.28
	1.210	1.213	1.215	2.217	1.220
60	28.75	28.68	28.63	28.58	28.51
	1.200	1.203	1.205	1.207	1.210
70	29.02	28.94	28.87	28.84	28.75
	1.189	1.192	1.194	1.196	1.200
80	29.26	29.19	29.14	29.09	29.02
	1.179	1.182	1.184	1.186	1.189
90	29.51	29.44	29.39	29.34	29.26
	1.169	1.171	1.174	1.176	1.179
100	29.79	29.72	29.66	29.61	29.51
	1.158	1.161	1.163	1.165	1.169
110	30.05	29.98	29.92	29.87	29.79
	1.148	1.151	1.153	1.155	1.158
120	30.32	30.23	30.18	30.13	30.05
	1.138	1.141	1.143	1.145	1.148
130	30.61	30.53	30.48	30.42	30.32
	1.127	1.130	1.132	1.134	1.138
140	30.88	30.80	30.75	30.70	30.61
	1.117	1.120	1.122	1.124	1.127
150	31.17	31.08	31.02	30.97	30.88
	1.107	1.110	1.112	1.114	1.117
160	31.45	31.36	31.31	31.25	31.17
	1.097	1.100	1.102	1.104	1.107
170	31.77	31.68	31.62	31.56	31.48
	1.086	1.089	1.091	1.093	1.096
180	32.06	31.98	31.91	31.86	31.77
	1.076	1.079	1.081	1.083	1.086
190	32.36	32.27	32.21	32.15	32.06
	1.066	1.069	1.071	1.073	1.076
200	32.70	32.61	32.52	32.45	32.36
	1.055	1.058	1.061	1.063	1.066
210	33.01	32.92	32.86	32.80	32.70
	1.045	1.048	1.050	1.052	1.055
220	33.37	33.27	33.20	33.14	33.01
	1.034	1.037	1.039	1.041	1.045
230	33.69	33.59	33.52	33.46	33.37
	1.024	1.027	1.029	1.031	1.034

sponding to atmospheric pressure at sea level) *from* water at 212°F. The factor for reducing the weight of water *actually* evaporated from the temperature of the feed, at the observed steam pressure, to equivalent evaporation under standard conditions is called the *factor of evaporation.* In Table XVI are given factors and also the actual evaporation per boiler horse power.

The factor is,

$$F = \frac{H - q}{970.4} \tag{81}$$

where H = total heat in steam above 32°F. for the initial steam pressure (from steam tables), q = total heat in feed water above 32°F., and 970.4 is the heat of vaporization of 1 lb. under standard conditions.

Take the case of boiler pressure 200 lb. gage and feed-water temperature 60°F.,

$$F = \frac{1199.2 - (60 - 32)}{970.4} = 1.207$$

which may be read directly from Table XVI. This table also gives the actual evaporation, equivalent to 1 boiler horse power.

Equation 81 also applies to superheated steam, using Table XIV for total heat. If feed water is 60°F. and steam 200 lb. and 200°F. superheat,

$$F = \frac{1309.7 - (60 - 32)}{970.4} = 1.32.$$

It would, therefore, require 1.32 times as much heat to vaporize 1 lb. of water at 60°F. into steam at 200 lb. and superheat it 200° as it would to vaporize 1 lb. at 14.7 lb. and 212°.

Fire-box and Tube Evaporation.—While all boilers are rated on the basis of their evaporation per square foot of heating surface, it should be noted that the values are merely averages for the entire surface. In a locomotive, the unit evaporation in the fire-box may be 55 or 60 lb., while at the front end of the boiler tubes it may be but 4 or 5 lb. During the Coatesville locomotive boiler tests of 1912, determinations were made of the fire-box and tube evaporation, results being shown in Fig. 62. The boiler was divided into two compartments, (*a*) fire-box and (*b*) tubes, water being weighed as it entered each. The evaporation when the boiler was developing its maximum power is represented at B, and at this point 13,500 lb. per hour were evaporated by the

fire-box and 3000 lb. by the tubes. Fire-box heating surface (including arch tubes) was 246 sq. ft.; tube surface, 3009 sq. ft.

$$\frac{13{,}500}{246} = 54.8 \text{ lb. per square foot, for fire-box.}$$

$$\frac{30{,}000}{3009} = 9.97 \text{ lb. per square foot, for tubes.}$$

The fire-box surface, which was but 7.5 per cent. of the total heating surface, evaporated 40 per cent. of the total water, which brings out the influence of high fire-box temperature, with its direct radiation to the metal over the hot gases, in producing high

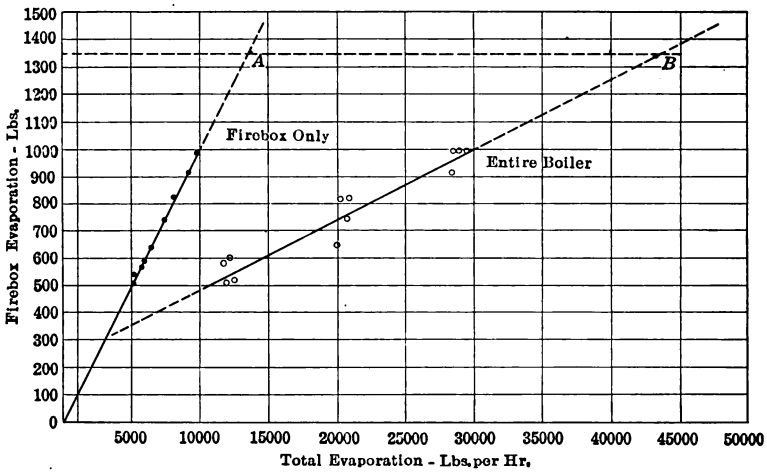


FIG. 62.—Relation of fire-box and tube evaporation, Coatesville tests.

evaporation rates. At a point corresponding to one-half the maximum evaporation, the fire-box evaporated 36.2 per cent. of the total.

By dividing the boiler into the two compartments for these tests, the normal circulation in a locomotive boiler was doubtless affected but based on these and other investigations, our present knowledge leads to the belief that from 35 to 40 per cent. of the total evaporation normally takes place in the fire-box.

A study of many records of temperature drop through tubes and flues indicates that the unit of evaporation of water surrounding the superheater flues is practically the same as it is about the small tubes, thus simplifying the problem of calculating the rela-

tive amount of heat transferred across the saturated and the superheated steam surface in superheater locomotives.[1]

Heat Transmitted to the Steam.—Figure 54, page 147, suggested a study which may be extended along somewhat different lines. Nussells' formula[2] for heat transmission has been found by the author to give a rational solution of this involved problem, but it should be used with the caution that no formula has more than limited application, and further that the data must be accurate and complete if the results are to be of value.

$$U = 0.0255\frac{K}{d}0.214\left(\frac{Wc_p}{AC}\right) \qquad (82)$$

U = transfer rate in B.t.u. per hour per square foot of surface of one tube per degree of difference in temperature.

W = weight in pounds of gas flowing through the tube per hour. May be calculated from the sum of the weights of air and the products of combustion.

A = inside area of tube, square feet.

c_p = specific heat of gas at constant pressure.

d = inside diameter of flue in feet.

C = conductivity of gas at the mean temperature and pressure in B.t.u. per hour per square foot of surface per degree Fahrenheit drop in temperature per foot. For air, this equals $0.0122(1 + 0.00132T)$; for CO_2 is $0.0076(1 + 0.00229T)$, where T is the mean temperature in Fahrenheit degrees at which the heat is conducted through the metal.

K = conductivity of the steam at the temperature of the wall of the flue; for superheated steam it equals $0.0119(1 + 0.00261T)$.

It is difficult to obtain the data from which to calculate results based on Eq. 82, the final values for evaporation usually being established from tests. Since tests have proved that the average evaporation per square foot of surface exposed to the superheater flues (located as shown in Fig. 84, of the following chapter) is practically the same as the unit of evaporation for the boiler tubes, it follows that by omitting one-fifth of the total surface, which would have been water heating surface and replacing it by

[1] The quantity of heat transferred is greatly affected by the condition of the flues. Tests on the Illinois Central Railroad, reported in the *Railroad Gazette*, Jan. 27, 1899, showed an increase of 13 per cent. in evaporation when $\frac{3}{64}$ in. of scale had been removed.

[2] "Steam," The Babcock & Wilcox Co., New York.

an equal area exposed to the hot gases in the flues, the evaporation of the boiler as a whole remains practically the same. On the other hand, the transfer rate (based on equivalent evaporation) across the superheating surface is not over 0.5 to 0.7 of that across the saturated steam surface.

In tabulating general dimension of a superheater applied to a locomotive (as Table XXII, Chap. XII), it is common to give the *equivalent heating surface* as the total evaporative heating surface + 1.5 times the superheating surface, as noted on page 160. In view of the statement in the previous paragraph, this is manifestly inconsistent, as far as the boiler alone is concerned, since the unit transmission to superheated steam is *less* than the unit transmission to saturated steam. Mr. H. H. Vaughan of the Canadian Pacific, who introduced this factor of "1.5," states that it was intended merely to approximate the heating surface which a boiler of the same size would have if designed without superheating surface; in other words, it is purely a factor for comparing the size of two boilers, one of which was equipped with a superheater, the other being of the ordinary type.

Problem 24.—A locomotive boiler evaporates 20,000 lb. of water an hour with gage pressure 180 lb., barometer reading = 30 in., feed-water temperature, 50°F.; find the equivalent evaporation. Compare this result with a superheater locomotive evaporating the same total amount of water and working on 180-lb. gage; atmospheric pressure 14.5 lb.; superheat 250°F.

STEAM ACTION

Definitions.—*Isothermal lines* represent constant temperature conditions. In the case of a gas, the curve of isothermal expansion is a rectangular hyperbola, having the equation:

$$PV = \text{constant} \tag{83}$$

The construction of this expansion curve is explained in reference books on steam engines. The work done during isothermal expansion from volume

$$V_1 \text{ to } V_2 = W = \int_{V_1}^{V_2} PdV = P_1V_1 \log_e \frac{V_2}{V_1} \tag{84}$$

The expansion of volume which occurs during the conversion of water into steam under constant pressure is *isothermal.* Such lines for saturated steam are lines of uniform pressure, as line 2–3, Fig. 63; for superheated steam the lines are rectangular hyperbolas, since steam, highly superheated, acts like a perfect gas.

Adiabatic lines represent a change in pressure and volume without loss or gain of heat by conduction, or radiation or internal chemical action. If P_1V_1 is the higher and P_2V_2 the lower pressure and volume, then:

$$P_1V_1^n = P_2V_2^n; \frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^n; \frac{V_1}{V_2} = \left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$$
$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}; \frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{n-1} \qquad (85)$$

where "n" must be determined for the conditions in the problem. Geometrically, n determines the slope of a line following the law expressed by terms in Eq. 85. The work during adiabatic expansion (or compression)

$$W = \int PdV = \frac{P_1V_1 - P_2V_2}{n-1} \qquad (86)$$

The actual expansion line of saturated steam is nearly that of the isothermal, while the expansion line for superheated steam is an adiabatic with the mean value of "n" about 1.35.

The *real or actual cut-off* is the ratio of the volume up to cut-off including clearance, to the total cylinder volume, including clearance.

The *apparent cut-off* is the ratio of the volume represented on the indicator card up to cut-off, to the volume of piston displacement. The actual cut-off is, therefore, a longer one. The cut-off is the reciprocal of the ratio of expansion. If k = clearance, expressed as a decimal of the volume displaced by the piston in one stroke and m is the apparent cut-off, then the actual ratio of expansion is $\frac{1+k}{m+k}$.

Compression in the clearance space uses that steam to advantage in addition to "cushioning" the effect of inertia of the reciprocating parts. In Fig. 66, if it is desired to produce a compression pressure of p, then the fraction of the stroke at which exhaust should close to produce this pressure is, approximately:

$$\frac{cb}{ca} = 1 - \left(\frac{p}{p'} - 1\right)k$$

where k is clearance, as above explained.

Steam Cycles.—In the *Carnot cycle,* the gas expands first isothermally and then adiabatically, is then compressed iso-

thermally and finally closing the cycle by being compressed adiabatically. It represents ideal conditions and may be studied with the view of increasing the efficiency of an engine. The Carnot cycle for steam is shown by the lines 2–3–4–5–2, Fig. 63. The efficiency of the Carnot cycle is expressed by

$$E = \frac{T_1 - T_2}{T_1} \tag{87}$$

where T_1 and T_2 are the higher and the lower absolute temperatures of the working substance. It is an ideal cycle and has no direct application in the study of practical locomotive operation. The value of E in an engine for saturated steam varies according to the initial and back pressures; for 200 lb. absolute, expanding to 20 lb. absolute (5.3 lb. gage) it is 18.3 per cent. The actual thermal efficiency is not over a half of the above under similar pressure conditions.

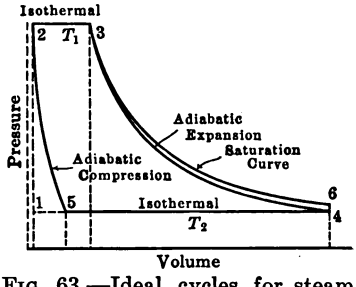


FIG. 63.—Ideal cycles for steam not superheated.

The Rankine cycle, also known as the Clausius cycle, is the common reference diagram for comparing the action of steam engines. It is shown by line 2–3–4–1–2, Fig. 63. The Rankine dry steam cycle 2–3–6–4–1–2 assumes that the steam remains dry saturated during expansion. Individual conditions determine which of the above named approach nearest to the expansion as shown on the actual indicator card; the former is the most satisfactory as a reference cycle. A modification of the dry steam cycle takes account of the heat remaining in the cushion steam at the end of each stroke.

The Rankine cycle has four stages:

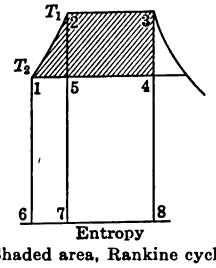
1. Feed water raised from temperature of exhaust to temperature of admission steam.
2. Evaporation at constant admission temperature.
3. Adiabatic expansion down to back pressure.
4. Rejection of heat at constant temperature corresponding with the back pressure.

These stages are represented in the entropy-temperature diagram, Fig. 64, each point of which has been lettered to correspond with similar points on the pressure-volume diagram.

The Rankine dry steam cycle differs from the one just described, (1) by addition of heat to the cylinder to maintain dry steam throughout the period of expansion; (2) by ending the expansion at some pressure higher than the back pressure, as shown in Fig. 63.

Entropy-temperature Diagrams.—A brief explanation is here given of the construction and application of these important diagrams.

In the indicator card, areas represent *work* in foot-pounds, while in the entropy-temperature or heat diagram, areas under the corresponding lines (down to absolute zero) represent *heat* expressed in B.t.u. or its equivalent. Select any small square to represent a B.t.u.; then will one side represent 1°F. difference in temperature and the other side a quotient which is called *entropy*. By establishing the number of B.t.u. in a square inch,



Shaded area, Rankine cycle.

FIG. 64.—Entropy diagram.

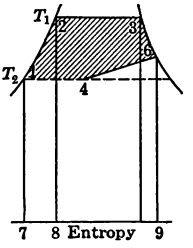


FIG. 65.—Rankine dry steam cycle.

or in a small square on cross-section paper, it is possible to determine the heat absorbed and delivered for the different operations represented on the indicator card, by measuring the area under the diagram.

The simplest form of an entropy-temperature diagram is shown in Fig. 64. Line 1–2 represents the process of raising the temperature of the liquid from 32°F. to a temperature T at which it is evaporated into steam. This line may be laid off by plotting from Steam Tables entropy values for water corresponding to temperature points, from 2–3, during the evaporation of the water, the temperature remains constant but the entropy increases and therefore the line is horizontal.

From 3–4 is adiabatic expansion—a vertical line extending down to release. Exhaust is represented as taking place along 4–1. The area 6–1–2–7–6 represents in B.t.u. the heat of the

liquid; area 7–2–3–8–7 the heat of vaporization; the area 1–2–3–4–1, the work done.

$$\text{Efficiency} = E_1 = \frac{\text{area } 1\text{–}2\text{–}3\text{–}4\text{–}1}{\text{area } 6\text{–}1\text{–}2\text{–}3\text{–}8\text{–}6} \text{ for Rankine cycle}$$

$$\text{Efficiency} = E_2 = \frac{\text{area } 5\text{–}2\text{–}3\text{–}4\text{–}5}{\text{area } 7\text{–}2\text{–}3\text{–}8\text{–}7} \text{ for Otto cycle}$$

If the expansion is not adiabatic but follows the dry steam cycle, the condition is represented in Fig. 65, where the numbers correspond with similar points in Fig. 63. The efficiency is

$$E_3 = \frac{\text{area } 1\text{–}2\text{–}3\text{–}6\text{–}4\text{–}1}{\text{area } 7\text{–}1\text{–}2\text{–}3\text{–}6\text{–}9\text{–}7} \text{ for dry steam cycle}$$

A typical indicator card is shown in Fig. 66. The principal

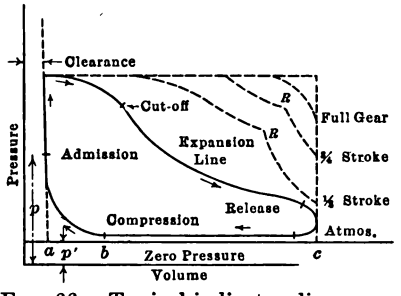
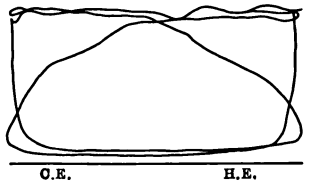


FIG 66.—Typical indicator diagram.



Steam chest diagram shown above cards

FIG. 66*a*.—Low speed cards—13.9 m.p.h.

conditions which cause an actual indicator card to differ from the theoretical work diagram, may be summarized as follows:

1. Incomplete compression of the cushion steam.
2. Wire-drawing in admission.
3. The chilling action of the metal on steam causing cylinder condensation.
4. Incomplete expansion.
5. Radiation.
6. Leakage of steam.
7. In compound engines, additional wire-drawing in the transfer of steam from one cylinder into another.

These losses may be shown as areas on the entropy diagram and their percentage readily calculated and from which one may study practical methods of development to give higher efficiency. The indicator card *A–B–C–D–E–F*, Fig. 67, is from locomotive No. 1, of the Pennsylvania State College. The cylinders are

17 in. by 24 in. and boiler pressure 125 lb. The pressure and temperature scales shown to the left in the lower diagram are both in absolute units and points on the lower diagram are lettered to correspond with similar points on the indicator card. The pressure scale is added merely for convenience in the study and helps to fix in the eye those relations of pressure and temperature which are otherwise left entirely to the imagination. If there was no cushion steam left in the cylinder at the end of each stroke, E^1 would fall at G^1.

Loss (1) noted above, is represented in Fig. 67 by area $F^1NA^1F^1$; loss (2) by $A^1TB^1A^1$; (3) by $TRSC^1T$; (4) by $C^1PD^1C^1$. The card $A^1B^1C^1D^1E^1F^1A^1$, as finally corrected, is found to have but 65 per cent. of the area of the Rankine cycle $MRUG^1M$.

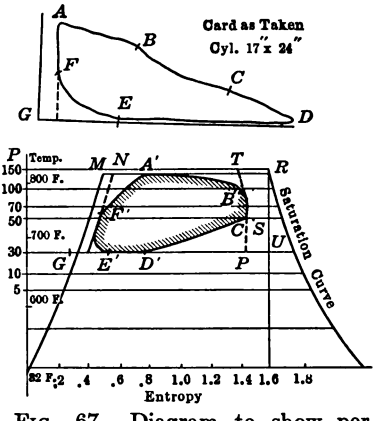


FIG. 67.—Diagram to show percentage of actual to theoretical heat available.

In Eq. 11, Chap. III, the tractive force formula was written:

$$T.F. = \frac{0.85Pd^2s}{D} \times \text{speed factor}$$

It was pointed out that this speed factor depends upon the mean effective pressure, which becomes less as the reverse lever is "hooked up." In starting and at low speeds, the cut-off in full gear (see Fig. 66) gives nearly a rectangular card and steam is not used expansively; as the speed increases, the engineman pulls the reverse lever toward the center of the quadrant, thus cutting off steam earlier in the stroke, thereby adjusting the amount of steam to meet the requirements of the load and speed. (See Fig. 66*a*.) With full throttle opening the regulation is determined by the position of the reverse lever. The condition is radically different from that of a stationary engine regulated by a throttling governor.

Fig. 68 shows a group of typical cards, taken from the right side cylinder of an Atlantic type locomotive for speed varying from 28 to 84 m.p.h. and all but two of these are for 35 per cent.

cut-off; cards 1 and 7 are at 30 and 25 per cent. respectively. Above each card there are two numbers and a letter "F"; thus,

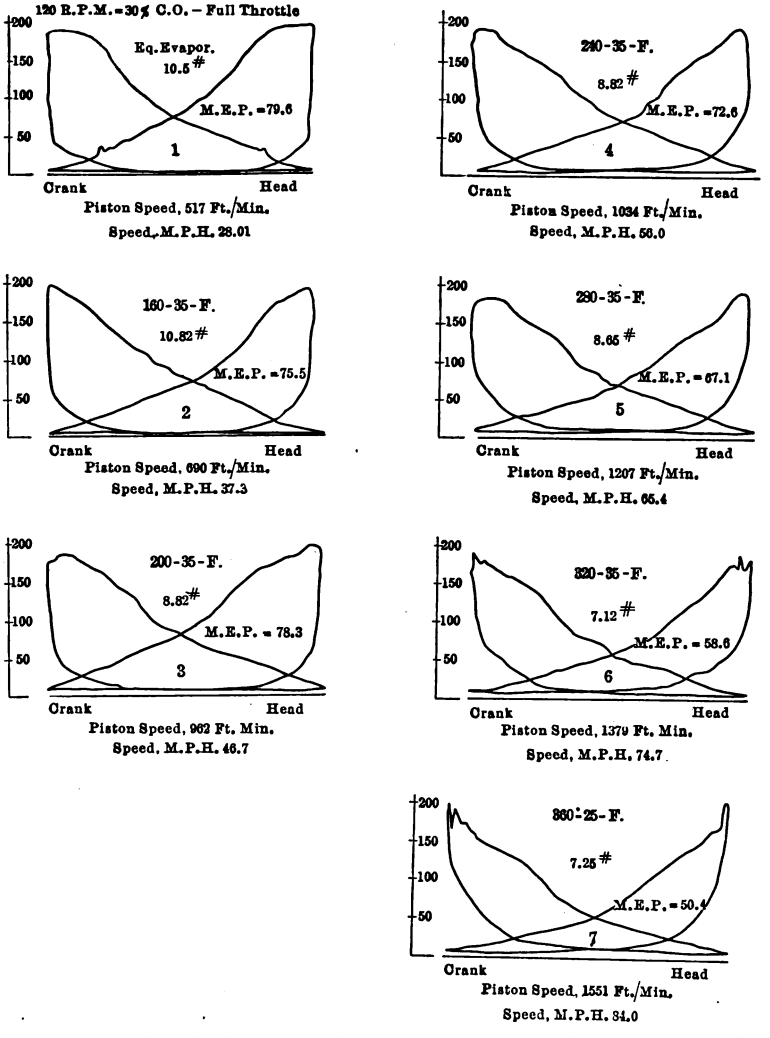


FIG. 68.—Typical indicator cards showing effect of increasing speed—4-4-2 type, E3sd class.

for card 2, 160–35–F means the speed was 160 r.p.m., 35 per cent. cut-off and full throttle opening.

Note the almost uniform expansion line to the end of the stroke. The release occurs in most of the cards at about 50 lb. pressure.

The indicated horse power of cards 5, 6 and 7 are the same within 10 hp. This shows that for these speeds, the mean effective pressure decreased in direct ratio to the increase of speed, which result may be checked from values given on the cards. Note that the cut-off pressure becomes less as the speed increases and compression at the end of the stroke increases at the higher speeds.

Directly above the expansion lines is given the equivalent evaporation from and at 212°. In general, as the speed increases

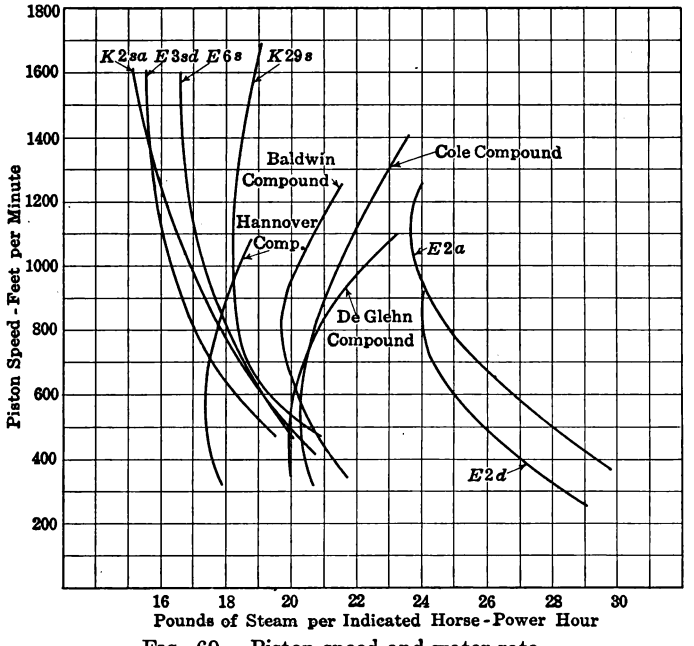


FIG. 69.—Piston speed and water rate.

the water rate decreases. A review of results at Altoona covering tests under various piston speeds, points to the conclusion that in simple expansion engines, both saturated and superheated water rate decreases as the piston speed increases, for speeds above 400 ft. per minute. A summary of results for many types of locomotives, showing the relation of piston speed and water rate are shown in Fig. 69. The conclusion stated above for simple engines does not seem to apply to compounds. One of the advantages of superheat, which this dia-

gram brings out, will be referred to in the next chapter. Note that the best water rate is low, between 15 and 15.5 lb., for the higher speeds for both the Atlantic and the Pacific types.

Referring again to Fig. 68, the better economy at the increased piston speed may also be influenced by the amount of superheat in the steam at the higher speeds. Note the values in the following table:

Card	Piston speed ft./min.	F.° superheat in branch pipe
1	517	138
2	690	199
3	962	179
4	1034	213
5	1207	226
6	1379	228
7	1551	221

At piston speeds above 1000 ft., the amount of superheat is nearly the same and by referring to the curve in Fig. 69 for the E3sd, we find it is nearly a vertical line above 1000 ft. per minute. The E2a saturated steam locomotive with cylinder $20\frac{1}{2} \times 26$ in. gives evidence of reaching a maximum economy at about 1100 ft. per minute, the general shape of the curve being the same as for the superheated engine. This engine was not influenced by superheat and undoubtedly warrants the conclusion that the water rate decreases because of increased piston speeds.

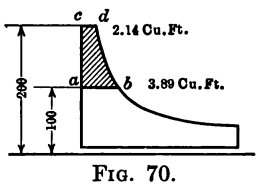


FIG. 70.

Effect of Pressure.—Note that in Table XV (or in Table XIII) as the pressure and temperature increase, the latent heat decreases; but this decrease being less rapid than the corresponding increase in the heat of the liquid, the total heat increases a small amount with the increase in pressure. For example, as the pressure of saturated steam is raised from 100 to 200 lb. the total heat in each pound is increased but 10.4 B.t.u., or less than 1 per cent. of the total heat. This very small addition in heat units is not sufficient to account for the large gain in work obtained by the higher pressure. The true advantage is brought out in Fig. 70. By raising the boiler pressure from 100 to 200 lb. an increase in work as represented

by the area *a–b–c–d–a* can be obtained in the theoretical diagram. This gain is obtained chiefly from two causes:

1. The volume of a pound at 200 lb. pressure is 45 per cent. less than it is at 100 lb. and therefore the same weight of steam does more work in expanding from the higher pressure and this additional amount of work requires but few additional heat units to raise the pressure.

2. From Eq. 87, the efficiency of a perfect heat engine is expressed by $\frac{T_1 - T_2}{T_1}$. The actual steam engine expansion with saturated steam is sufficiently near to that of the theoretical to warrant its use as an illustration. Suppose the exhaust, in the case considered, is at a temperature corresponding to atmospheric pressure, or 212°F., 212° + 461° = 673°F. absolute. For the 200 lb. pressure, $\frac{848 - 673}{848} = 0.206 = 20.6$ per cent. and for 100 lb. $\frac{799 - 673}{799} = 0.158 = 15.8$ per cent. The gain of 20.6 − 15.8 per cent., being theoretically gained by the higher pressure. By examining Fig. 70, it will be noted that the gain in work areas becomes relatively less as the pressure increases. The adoption of the superheated steam locomotive has introduced a problem of maximum allowable temperatures, more important than one of limiting pressures.

Professor Goss conducted an important research on the Purdue University testing plant to determine the steam economy from increased pressure, the significant results of which are shown in Fig. 71. The steam consumption for an engine working on the theoretical Carnot cycle is shown by the curve *C–D* and the test results for saturated steam by *A–B*. The curve *E–F* has been plotted for steam superheated about 150°F., also from tests on the Purdue plant. Comparing *A–B* and *E–F*, we find a steam saving by superheating of 15 per cent. over saturated steam at 180 lb. pressure. The curve *A–B* between the pressure 135 and 205 lb., checks well with results of tests at the Altoona plant on an H6b (saturated) engine under throttling conditions and at constant drawbar pull.

The Purdue tests have led to the following conclusions:

1. The evaporative efficiency of a locomotive boiler is but slightly affected by changes in pressure, between the limits of 120 lb. and 240 lb.

2. Changes in steam pressure between the limits of 120 lb. and 240 lb. will produce an effect upon the efficiency of the boiler which will be less than ½ lb. of water per pound of coal for all pressures between the limits of 120 lb. and 240 lb. with an average error for any pressure which does not exceed 2.1 per cent.

3. It is safe to conclude that changes of no more than 40 or 50 lb. in pressure will produce no measureable effect upon the evaporative efficiency of the modern locomotive boiler.

The effect of different pressures upon smoke-box temperatures was found to be as follows:

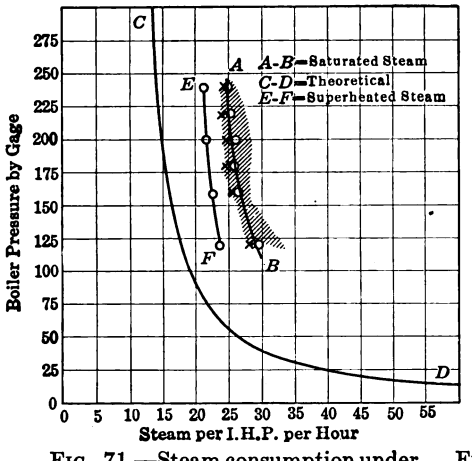


FIG. 71.—Steam consumption under different pressures.

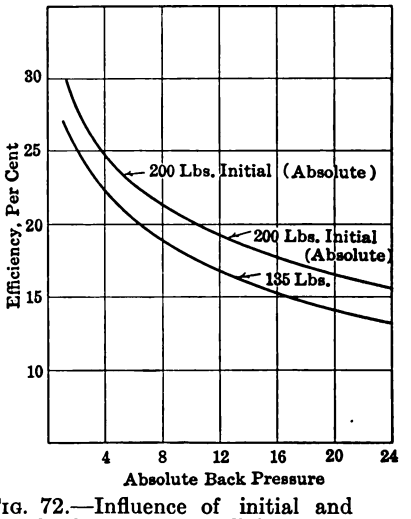


FIG. 72.—Influence of initial and back pressure on efficiency.

1. The smoke-box temperature falls between the limits of 590°F. and 850°F., the lower limit agreeing with a rate of evaporation of 4 lb. per foot of heating surface per hour and the higher with a rate of evaporation of 14 lb. per foot of heating surface per hour.

2. The smoke-box temperature is so slightly affected by changes in steam pressure as to make negligible the influence of such changes in pressure for all ordinary ranges.

The shaded zone, Fig. 71, represents the range of performance as it appears from all tests run under the several pressures employed. It shows that the variation in performance for all conditions of running which are possible with a wide open throttle

scarcely exceeds 5 lb. For purposes of comparison, it is desirable to define the effect of pressure on performance by a line, and to this end an attempt was made to reduce the zone of performance to a representative line. In preparing to draw such a line, the average performance of all tests at each of the different pressures was obtained and plotted, the results being shown by the small circles on curve A-B.

Initial Pressure.—Initial pressure in the cylinder becomes lower as the speed increases. Henderson gives the following values:

TABLE XVII.—SPEED AND INITIAL PRESSURE

Revolutions per minute	0	50	100	150	200	250	300	350
Initial to boiler pressure	0.98	0.95	0.92	0.90	0.88	0.87	0.86	0.85

Table XXVII, Appendix, gives the revolutions per minute for various diameters of driving wheels and speeds.

Back-pressure.—Following the consideration of the influence of different initial pressures, we may inquire into the effect of varying the back-pressure. This problem is somewhat involved, since changing the back-pressure alters the drafting conditions and thereby alters the operation of the locomotive as a whole.

Theory gives the results presented in Fig. 72, which shows that the cylinder efficiency is notably increased by decreasing the back pressure. The cards in Fig. 68 show a decreasing amount of back pressure as the speed increases. Fig. 73 gives the results on four Atlantic type locomotives, comparing back pressure and indicated horse power. The E6s is a superheater locomotive similar in other respects to class E6 and the same applies to the E3d and E3sd, except that the diameter of the cylinders in the latter is 1½ in. larger than in the former. The boiler tubes of the E3sd are 9.4 per cent. longer than those of the E6s, which is favorable for less back pressure in the former, although this is not sufficient to account for the large decrease in back pressure in the E6s class. The reduction of back pressure to that found in the E6s is a noteworthy achievement in design, based upon test results and operating conditions. The increase in the least back pressure for the E2d ranges from 79 per cent. at 500 d.hp. to 82 per cent. at 1150 d.hp. above that obtained in the E3sd locomotive.

With these results in mind, study again Fig. 72. For example, at 200-lb. initial pressure the reduction in pressure from 24 down

to 18 lb. means an increase of about 2 per cent. in engine efficiency. From Fig. 73, the superheater locomotives would average about this amount of reduction in pressure for a considerable range in power output, and the increased efficiency of superheater locomotives must be due in part to reduction in back pressure.

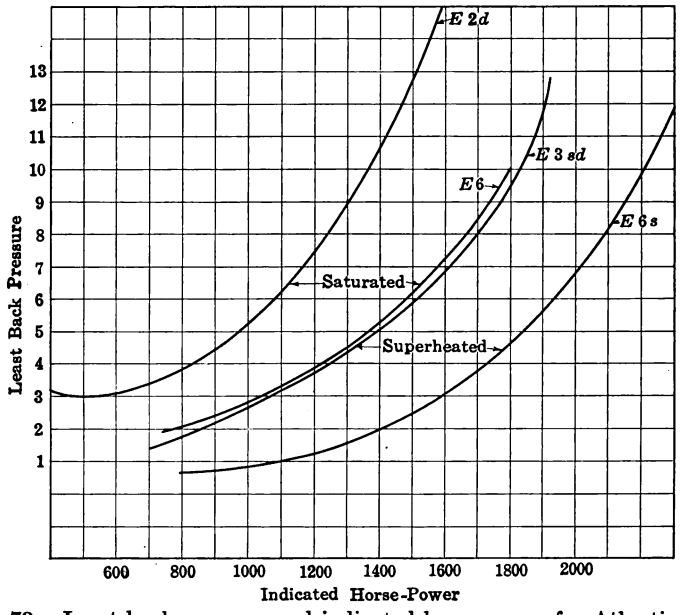


FIG. 73.—Least back pressure and indicated horse-power for Atlantic type engines.

At corresponding indicated horse-powers, the superheated steam locomotive exhausts with a low back pressure, about one-half that of the saturated steam locomotive.

Review Problem in Design

Problem 25.—Proportion the leading parts of the boiler for an Atlantic type locomotive to haul six 70-ton passenger cars up a 1 per cent. grade at 25 m.p.h.; drivers, 80 in. diameter; boiler pressure, 200 lb.

Proceed as follows for *approximate* results. Other methods for problems similar to this are given in Chap. XII. (*a*) From Chap. V, find the total train resistance in pounds. (*b*) Assume a factor of adhesion of 5.0 and find the weight on drivers to haul the required load, including the tender. For this problem take weight of tender and the proportion of weight on drivers to total weight of locomotive, from the values given for Atlantic type locomotive in Table II, Chap. II.

(*c*) Add to (*a*) the resistance of engine and tender, giving the total tractive force required.

(*d*) Determine the proper piston speed from

$$S = \frac{56Vs}{D} \tag{88}$$

S = piston speed, feet per minute.
V = speed of train, m.p.h.
s = piston stroke in inches; assume 28 in.
D = diameter drivers, inches.

In our case,
$S = \frac{56 \times 25 \times 28}{80} = 490$ feet per minute, which result may be read directly from Table XXVIII Appendix.

(*e*) From Eq. 11, Chap. III, find the diameter of the cylinder. To obtain speed factor, use Fig. 12.

(*f*) Find the pounds of water to be evaporated per minute.

$$W = 2vsnwf \times 1.25 \tag{89}$$

v = volume in cubic feet of both cylinders.
s = percentage of stroke at which cut-off takes place, expressed as decimal. (Take speed factor, as in Fig. 12, as the ratio of initial to mean pressure. Consult tables, as on p. 930 Kent's Hand Book, for cut-off corresponding to this ratio.)
w = weight of cubic foot of steam at cut-off pressure (steam tables).
f = factor of evaporation (Table XVI).
1.25 = factor for losses not otherwise considered.

(*g*) If bituminous coal, allow 1.5 lb. coal per square foot heating surface per hour and solve for area heating surface, sq. ft. = A.

$$A = \frac{E}{mw} \tag{90}$$

E = pounds water to be evaporated per hour.
m = pounds water evaporated per pound coal.
w = pounds coal per square foot heating surface.

For Pennsylvania bituminous coal, assume $m = 7$.

(*h*) Find grate surface, G, in square feet.

$$G = \frac{E}{mh} \tag{91}$$

where E and m are as above and h = coal burned per square foot grate per hour assume 100.

(*i*) Make fire-box heating surface (in this case) to contain 180 sq. ft. and find the amount left for tubes.

(*j*) Assume tubes 2 in. diameter and 18 ft. long and find the number required for the tube heating surface. (Table XXXII, Appendix.)

VALVE MOTION

A few years ago, the Stephenson link motion and the D-valve were universally used upon American locomotives. To-day, the Stephenson motion is rapidly being displaced by *outside* gears, chiefly the Walschaerts, and the old standard valve has given place to the flat balanced valve and the piston valve. These radical developments have come about largely through necessity, caused by the introduction of the heavier types of engines to be worked to the best economy. In dealing briefly with this subject, attention will be directed to the Walschaerts gear, the one now in general favor in this country, although it has been used on the continent quite generally since it was put into service in 1844, a year after the Stephenson gear came into prominence. We will discuss the principles of construction, which will also be found common to other radial valve gears.

The Stephenson gear is heavy, is placed under the boiler, where repairs cannot readily be made, and has large bearing areas, which must be kept well oiled. *From point of theory*, the advantage over the Walschaerts motion is the fact that the Stephenson has a variable lead which gives a greater preadmission of steam and this lead increases as the cut-off is made earlier.

The Walschaerts gear is lighter, can be more easily repaired, is more permanently adjusted, requires but a single eccentric return crank, transmits the moving force to the valve in nearly straight lines, and as a whole is simpler to understand. The gear of the future will probably be one which cuts off the steam more quickly than either of the two mentioned above.

Definitions.—*Outside Lap* is the amount the valve extends over the edge of the steam port, when the valve is in the center of its travel. It serves the purpose of increasing the expansion (Fig. 74). *Inside or Exhaust Lap* is the amount the exhaust side of the valve overlaps the port, causing the valve to exhaust the steam later in the stroke, producing greater compression.

Lead is the amount the valve has opened port when the crank pin is on either center, thus providing preadmission of steam before the piston has completed its stroke (Fig. 75).

Thus the two events of admission and cut-off are determined by the steam lap, and each begins when the valve is at the same distance from its central position. Whether the position is that corresponding to admission or to cut-off depends upon the

direction in which the valve is moving at the instant. Similarly the events of release and compression are determined by the exhaust lap and each event begins when the valve is at the same distance from its central position, the position fixing the event of release or compression according to the direction of motion of the valve.

These two positions may, therefore, be appropriately called the critical positions of the slide valve.

Negative Steam Lap is the width of the port opening on the steam side when the valve is in the center of its travel. *Negative Exhaust Lap* is the width of the port opening on the exhaust lap. The negative lap must be added to the distance of the valve from its central position to find either the opening for steam or exhaust. Negative steam lap is rarely used but negative exhaust lap is more common and may be employed in express passenger

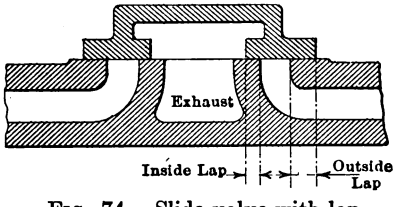


FIG. 74.—Slide-valve with lap.

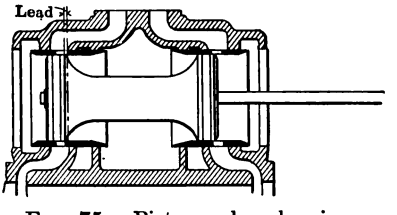


FIG. 75.—Piston valve showing "lead."

engines in order to prolong the period between release and compression at high speeds.

We may trace various steps in development of the Walschaerts motion, as outlined by C. J. Mellin, of the American Locomotive Company.

THE DEVELOPMENT OF THE WALSCHAERTS GEAR

Simplest Form of Valve Motion.—Figure 76 represents the simplest form of valve motion, which consists of a single eccentric of the return crank design, driving, by means of an eccentric rod directly connected to the valve stem, a plain D-valve without lap or lead.

When running forward, with the crank pin on the back center, as shown, the eccentric crank pin must be on the top quarter; in this case the valve will be in its central position with all ports closed. With the throttle open the left-hand engine, being con-

nected to the wheel on the other side of the axis with its main crank pin on the lower quarter, and the valve in position to admit steam in front of the piston, would start the engine forward. The eccentric crank on the right-hand end (the one shown) would then move forward, uncovering the back steam port and admitting steam behind the piston, and the engine would continue to run forward. With this gear it is evident that the engine will run only in the direction in which it is started.

This simple form of engine may be reversed by introducing a rod between the eccentric and the valve stem, pivoted at its center so that one end moves with and in the same direction as the eccentric and the other in exactly the opposite direction.

Such a construction is shown in Fig. 77, with the valve stem at the lower end of the link and the main pin in the upper corner as shown in diagram A; the eccentric will have moved the

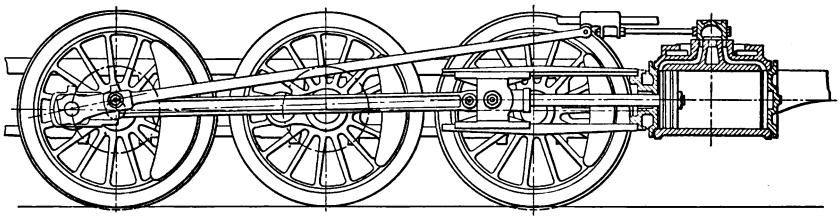


FIG. 76.—Simple form of valve motion.

valve to its extreme forward position and the back port will be open for the admission of steam behind the piston and the engine will run forward. If, however, with the main pin and eccentric crank in the same position, the valve stem was connected to the upper end of the link, as shown in diagram B, the valve would be moved to its extreme position to the left and the front port would be opened and steam admitted ahead of the piston and the engine would run backward. With a flexible valve stem, or radius rod, connected as shown, to a block which slides in a curved slot in the link, and with some suitable mechanism for raising and lowering the block, it is evident that the engine can be reversed.

This engine is far from being perfect; for, (1) as the valve does not close the port for the admission of steam until it is in its central position on the seat, and (2) since with the eccentric only a quarter of a revolution ahead of the main pin, it is not in this

position until the main pin is on the center, therefore steam will be admitted to the cylinders throughout the full stroke of the piston, which will cause the engine to use as much steam working against a light as against a heavy load. In order to govern the period of admission of steam (that is, give a variable cut-off), the valve must be redesigned so that it will close the steam port before it reaches its central position. This is accomplished by giving the valve "lap" and "lead," and then the valve motion must be so changed that the valve will be advanced from its central position on the seat a distance equal to the amount of the lap plus the lead, when the main pin is on either of the centers.

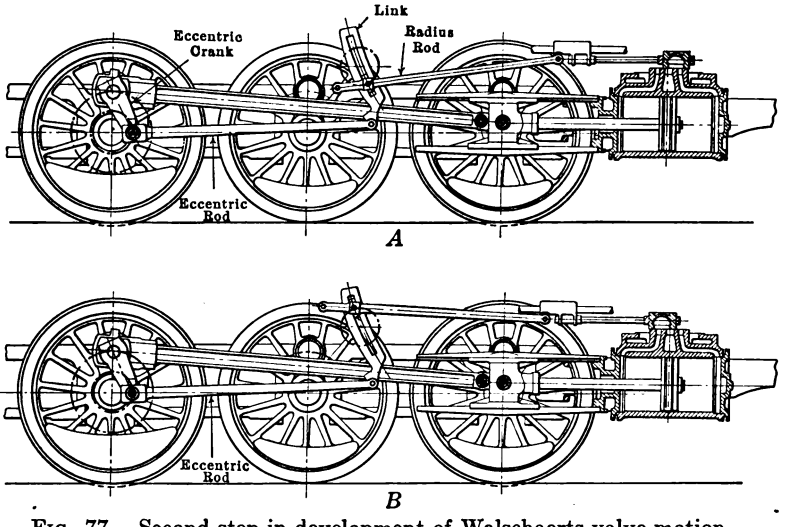


FIG. 77.—Second step in development of Walschaerts valve motion.

This advance of the valve cannot be obtained by any change in the position of the eccentric crank relative to the main pin. Fig. 78 shows the same valve motion as is shown in Fig. 77, except that the valve has 1 in. outside lap and the eccentric crank has been advanced to more than a quarter of a revolution ahead of the main pin. If then the link block is at the bottom of the link, as in position *A*, the advance given to the eccentric will have moved the valve forward a distance equal to the amount of lap plus the lead for the admission of steam, and the engine will run forward. If, on the other hand, the link block is moved to the upper end of the link, as shown in position *B*, the advance

given to the eccentric crank will have had the opposite effect, and the front port will be open and the reversibility of the engine thus *destroyed.* In order to provide lead, therefore, in any engine where there is but one eccentric, some means must be employed other than advancing or receding the position of the eccentric relative to the main pin, if there is to be any method of reversing the engine.

In the Walschaerts valve gear the motion for providing lap and lead to the valve is derived from the main pin by suitable connection with the cross-head.

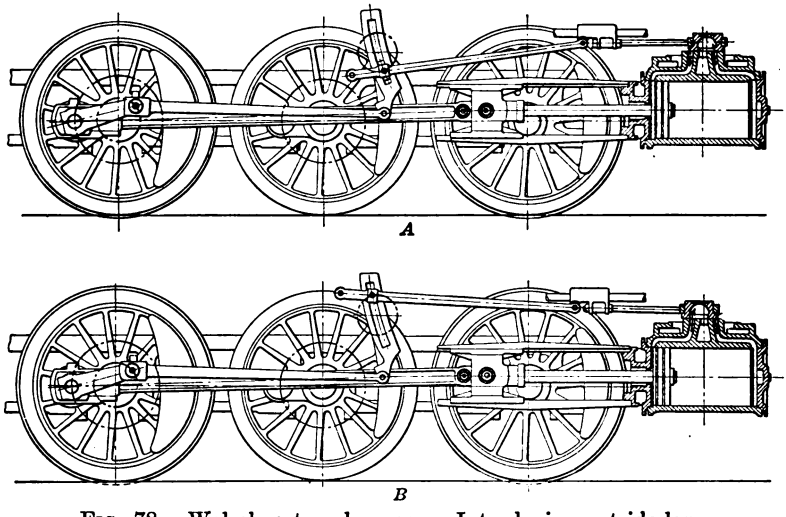


FIG. 78.—Walschaerts valve gear. Introducing outside lap.

Lap and Lead.—This introduces the next step in the development of the Walschaerts valve gear, the introduction of the "lap and lead," or "combination" lever. Suppose that the link block is in the center of the link as shown in Fig. 79, diagram *A*. As the center of the link block coincides with the center of the link support, there will be no movement of the radius rod, as the link swings back and forth through the action of the eccentric. If then the radius rod was connected to the combination lever at the point *R*, and the upper end of the lever was connected to the valve stem at *V*, and the lower end of the cross-head arm by means of a short link, as shown, as this latter moves back and forth, the point *R* being fixed, the point *V* will rotate

about it and the valve will be moved back and forth. With the main pin on the forward center, as shown in position *A*, the angle assumed by the combination lever has moved the valve back a sufficient distance to uncover the front port. With the main pin on the back center, as shown in position *B*, the combination lever is inclined in the opposite direction and the valve moved forward, and the back port is open. This is the way the Walschaerts valve gear derived its lead. The distance between the connecting points of the combination lever must be so proportioned that the travel of the cross-head from one end of the

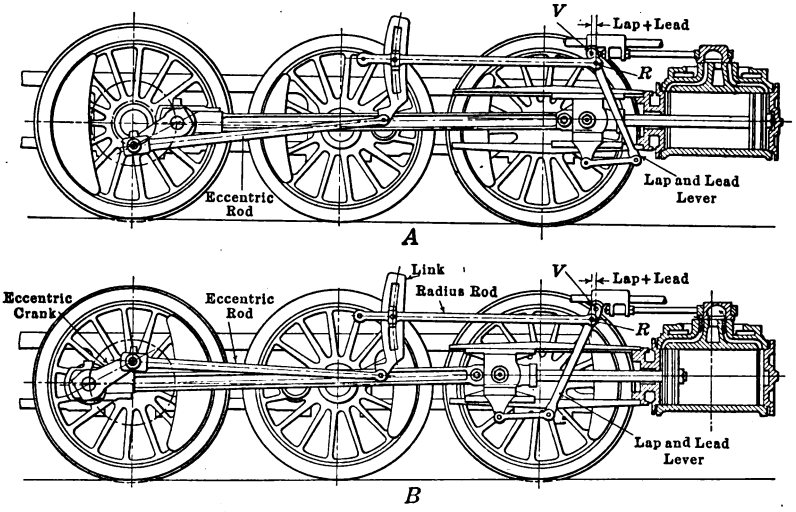


FIG. 79.—Introduction of "lap and lead," Walschaerts valve gear.

stroke to the other will give a travel to the valve equal to *twice* the lap plus the lead.

The position of the eccentric crank relative to the main pin depends on which end of the link is used for forward motion. If the forward motion is taken from the bottom of the link, the eccentric crank is a quarter of a revolution ahead of the main pin when the wheel is running forward if the valves have outside admission, and a quarter of a revolution behind it, if the valves have inside admission. In all cases, if the valves have outside admission, the radius rod is connected to the combination lever below the valve stem, and above it if the valves have inside admission.

To change the lead of the Walschaerts gear, it is necessary either (1) to change the lap of the valve, reducing it to increase the lead and increasing it to reduce the lead, in which case the cut-off will occur at later or earlier periods in the stroke, respectively; or (2) to change the lengths of the arms or distances between the connecting points of the combination lever. Increasing the distance between the radius rod connection and the valve stem connection to the combination lever would increase the lead; or shortening this distance would decrease the lead.

Figure 80 shows a design on which are given the names of the various parts.

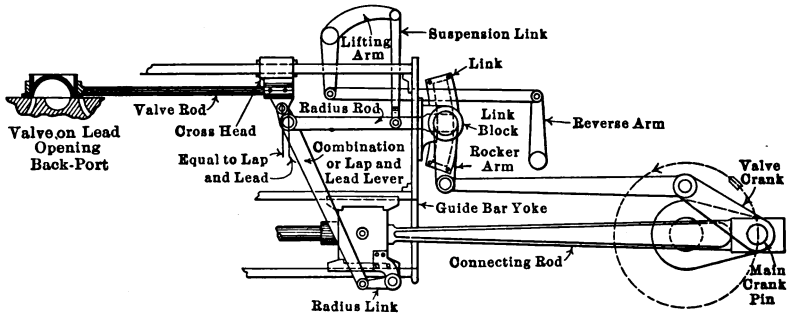


FIG. 80.—A design of the Walschaerts valve gear.

Theory.—Fig. 81 shows the gear at a dead point in dotted lines, and when the crank has moved angle θ in full lines.

Following Peabody's analysis of the gear, if the block d is at the middle of the link, the valve will derive motion from the cross-head only and the mechanism will be at mid-gear. The radius of the link arc is made equal to the length de of the radius rod; consequently the lead is constant for all settings of the gear. If the point h of the radius link hf were a fixed point, then the valve would receive motion from the eccentric OE, which has no angular advance. By placing the link block nearer the trunnion G, the motion is reduced; thus, motion from eccentric OE equals one-half as much if the block is halfway between d and G. If the link block is below G, the motion is reversed.

The displacement from mid-position of a valve moved by an eccentric, where θ is the crank angle and α the angular advance of the eccentric, is $e = r \sin(\theta + \alpha)$. Since the motion from the cross-head is equivalent to that from an eccentric 90° angular

advance (if motion is assumed harmonic), the valve derives a displacement from the cross-head of

$$e_1 = r_1 \sin (\theta + 90°) \mp r_1 \cos \theta$$

From the proportion of the lap and lead lever and the length R the crank, we have

$$r_1 = \frac{ae}{ef} R$$

The displacement of the valve from the influence of the eccentric OE is

$$e_2 = r_2 \sin (\theta + 0°) = r_2 \sin \theta$$

in which

$$r_2 = OE \frac{dG}{GF} \times \frac{af}{ef}$$

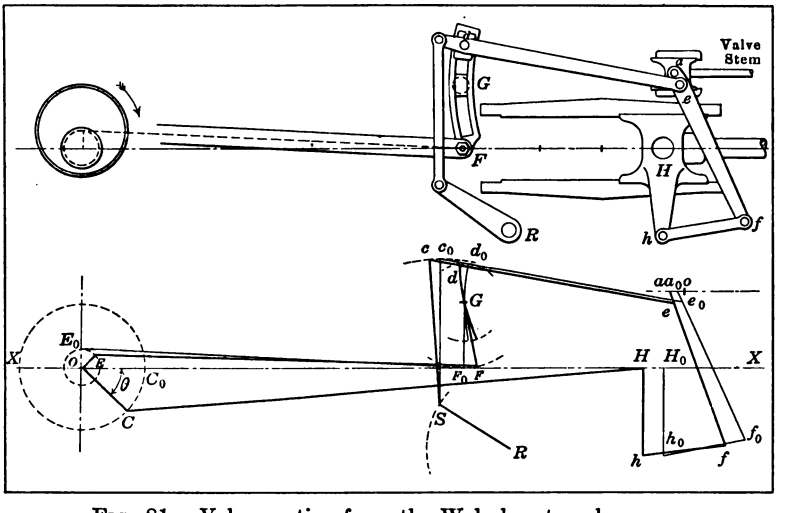


FIG. 81.—Valve motion from the Walschaerts valve gear.

The entire displacement e of the valve at any crank angle is the sum of the two independent displacements; therefore

$$e = e_1 + e_2 = r_1 \cos \theta + r_2 \sin \theta \qquad (92)$$

If, in Eq. 92, we let A and B be the coefficients of the trigonometrical values, we have

$$A = r_1 = \frac{ae}{ef} R \qquad (93)$$

and

$$B = r_2 = OE \frac{dG}{GF} \times \frac{af}{ef} \qquad (94)$$

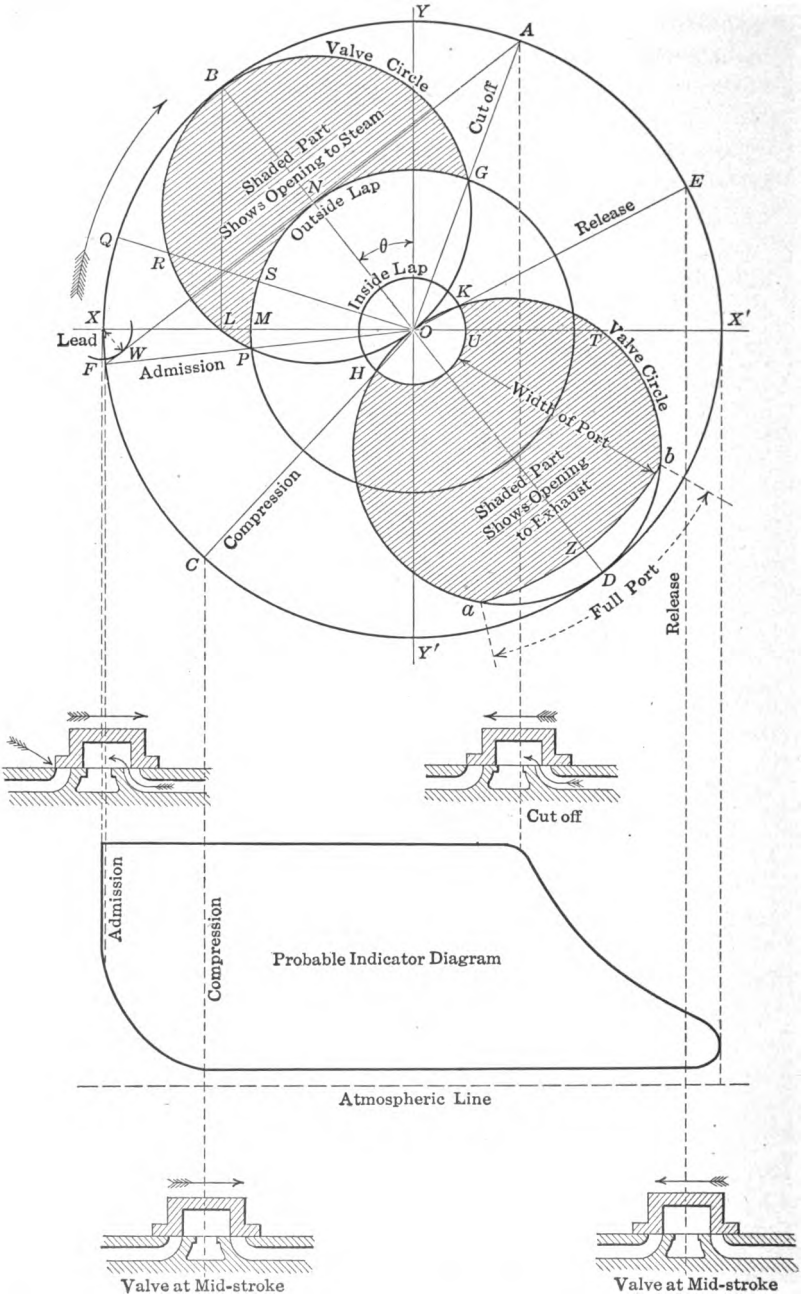


FIG. 82.—Zeuner's diagram of valve motion; also showing position of simple slide valve and distribution of steam in cylinder.

which will be the coördinates of the several valve circles in constructing a Zeuner diagram for this gear. As the connecting rod is not infinitely long, the diagram will contain slight errors.

Figure 82 presents the characteristics of the Zeuner valve diagram. With the radius equal to the half-travel of the valve, describe the circle *ABCDE*. Draw the axes *XX'* and *YY'*. Draw *OA* to represent the position of the crank at cut-off. With

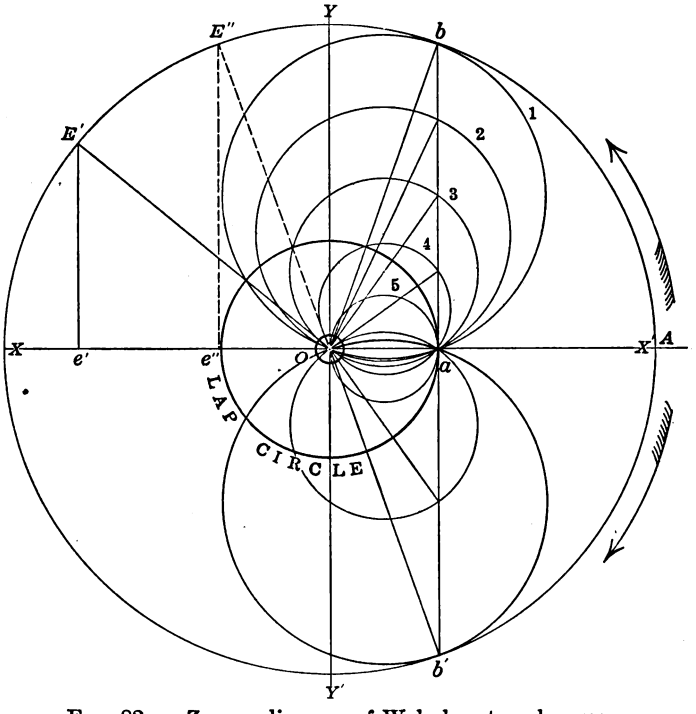


FIG. 82*a*.—Zeuner diagram of Walschaerts valve gear.

X as a center, and radius *XW* equal to the lead, describe part of the circle as shown, and from *A* draw *AF* tangent to this arc. Through *O* draw *BD* at right angles to *AF*. On *OB* and *OD*, describe valve circles and draw the *outside lap* circle as shown.

If the position of the crank at the point of compression or release is given, draw *OC* or *OE* and where this line cuts the valve circle, describe a circle with center *O* to pass through that point and the radius of this circle gives the amount of *inside lap*.

Therefore, when the crank is in the position OF, the valve is just beginning to open the steam port and when it reaches OX, the steam port is open an amount of the lead $= LM = XW$. When the crank reaches OE, the valve has passed the middle position and is distant from it on the other side an amount equal to OK, where release takes place. As the valve passes position OE, the valve continues to open the port to exhaust. Thus when the crank is at OX', the valve has moved from its central position a distance equal to OT and since OU is the inside lap, the port is open to exhaust represented by $OT - OU = UT$.

The position of the valves and the probable indicator card, as shown, will aid in making clear the application of the diagram in design.

From this diagram, the distance the valve has travelled may be determined for any position of the crank and consequently the opening of the port to steam for that point.

If the crank is in the position OQ and moving in the direction of the arrow, then will the distance which the valve has moved from its central position be given by OR and since the outside lap of the valve is equal to OS, the opening of the port to steam equals RS. When the crank reaches the position OB, the port is at its maximum opening equal to BN and after passing this point, the valve gradually closes the steam port, cutting off at G.

The application of this diagram to the Walschaerts valve gear will be apparent from Fig. 82*a*, which gives a Zeuner diagram from Henderson's "Locomotive Operation" of a valve motion of the Walschaert, type, having the lap 1 in., with no lead in full gear, the crank radius 12 in., and eccentricity of eccentric 3½ in. The arm GF of link is 8 in. and the extreme distance of block from fulcrum dG, 6 in. The combining lever is 26 in. total length, divided into a 2-in. portion and a 24-in. part. Lay off the rectangular axes $X - X'$ and $Y - Y'$, with the origin at O, as before.

From Eq. 93,

$$A = \frac{ae}{ef} R = \frac{2}{24} \times 12 = 1$$

and, as these values are all constant, the line b–b', erected on X–X' at point a, 1 in. from O, if made straight and perpendicular to X–X', will fix the ends of the diameters of all possible valve circles.

Equation 94 gives the value of

$$B = OE \frac{dG}{GF} \times \frac{af}{ef},$$

which for the greatest travel of valve becomes

$$B = 3\tfrac{1}{2} \times \frac{6}{8} \times \frac{26}{24} = 2.83 \text{ in.},$$

and which distance is laid off on the perpendicular above described from a to b and b'. The circles described upon the lines Ob and Ob' as diameters, are the valve circles for full travel of the valve, found to be 6 in. Equation 94, shows that the value of B depends directly upon the distance dG of the block d from the trunnion G and by dividing ab into four equal parts we obtain five valve circles. We see from these circles that no lead has been gained. Therefore, "hooking up" decreases the travel but does not alter the lead.

Practical experience in shop methods is necessary to adjust the Walschaerts gear for a particular engine. Theory merely points to the way.

Suggestions for Design.—The radius rod should be as long as it can be conveniently placed, at least eight times, or better, ten or twelve times the length of the travel of the link block; and, of course, the radius of the link must be equal to the length of the radius rod.

The shortest length of the eccentric rod should never be less than three and one-half times the throw of the eccentric.

If it will not shorten the effective throw of the link too much, the connection from eccentric rod to link should be brought down as near as possible to the center line of motion of the main rod.

The length of the long division of the lap and lead lever should not be less than two and one-quarter times the stroke.

The length of the lap and lead lever must be taken to suit the conditions under consideration in each case, so that the angle through which it oscillates will not exceed 60°, but less is preferable. The required horizontal movement or travel of the connecting point e of the radius rod to the lap and lead lever for a given maximum valve travel may be found by the formula:

$$b = \frac{R\sqrt{a^2 - c^2}}{R + c} \text{ for outside admission.} \qquad (95)$$

$$b = \frac{R\sqrt{a^2 - c^2}}{R - c} \text{ for inside admission.} \qquad (96)$$

R = radius of main crank.
c = lap and lead.
a = one-half the travel of the valve.
b = one-half the travel of the point e.

Valve Displacement Diagrams.—Figure 83 is from a set of curves prepared for the author by Mr. G. B. Wharen, a graduate of the Pennsylvania State College. It compares the valve displacement of a Walschaerts and a Stephenson gear with outside admission valve for the locomotive, P. S. C. No. 1, with the reverse lever in notches 1, 5 and 9 of the quadrant.

In each drawing, curves 1 and 3 are the valve displacement curves for the Walschaerts and the Stephenson gears respectively, laid off with equal crank angles as abscissæ, and actual valve displacement as ordinates. Curve 2 represents the piston displacement laid off for equal crank angles, as the base line; the ordinates are in per cent. of stroke.

To construct the displacement curves, first sketch in outline the gear, in a certain position (say on head end dead center); the position of the reverse lever fixes the position of the link block. A series of points or positions of the valves is then to be found for twelve equal crank angles. That is, place the crank at an angle, say of 30° in the forward motion, and draw in the gear as it would naturally be when the crank pin has assumed this position. We can then determine by measurement the amount the valve has moved from its mid-position, and so on around for the 360°.

On a horizontal axis step off the twelve equidistant crank angles. The twelve points which determine the valve displacement curves are measured on the valve stem from the mid-position to each position corresponding to its respective crank angle and laid off on ordinates through its respective crank position, perpendicular to the base line.

This was done for both motions, which set forth, probably for the first time, certain characteristics of the gears.

The condition of constant lead in the Walschaerts valve gear is brought out in these diagrams. Note that for each position the Stephenson gear opens up the steam port wider and the cut-off is later than in the Walschaerts motion. This, however, does not imply that the engineman has better control of the steam as a whole in the Walschaerts valve gear; rather, at high speeds, the opposite is more nearly the case.

With the Walschaerts valve gear the valve is given a rapid motion toward the cut-off position. For when cut-off occurs, the crank is somewhere near the quarter position and imparting its quickest motion to the cross-head, which in turn imparts a

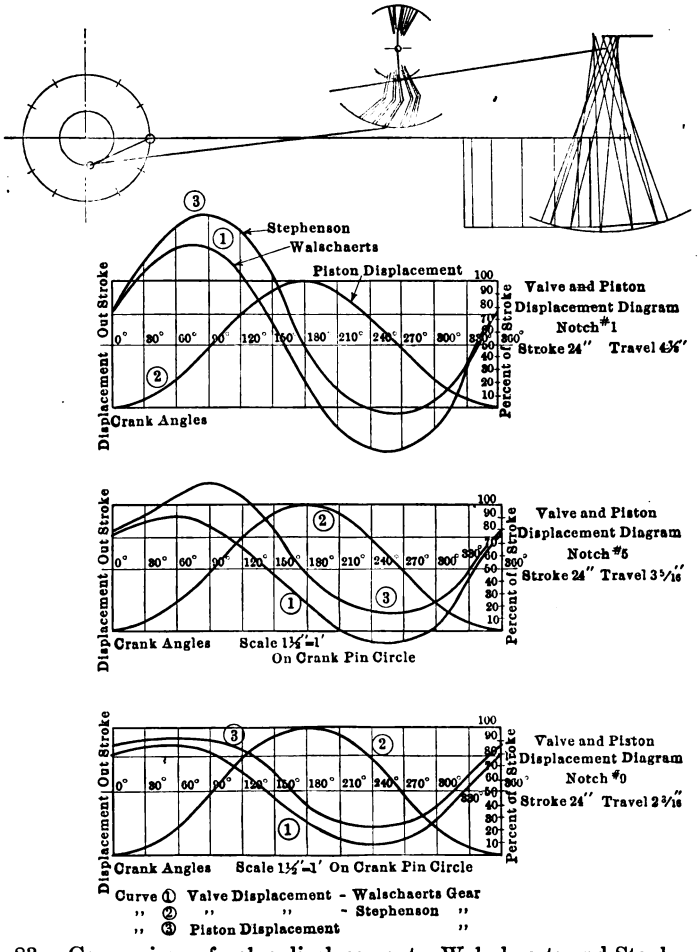

Stephenson
Walschaerts
Piston Displacement
Valve and Piston Displacement Diagram
Notch #1
Stroke 24" Travel 4 3/16"
Displacement Out Stroke
Percent of Stroke
Crank Angles
Valve and Piston Displacement Diagram
Notch #5
Stroke 24" Travel 3 5/16"
Crank Angles Scale 1½"=1' On Crank Pin Circle
Valve and Piston Displacement Diagram
Notch #9
Stroke 24" Travel 2 3/16"
Crank Angles Scale 1½"=1' On Crank Pin Circle
Curve ① Valve Displacement - Walschaerts Gear
" ② " " - Stephenson "
" ③ Piston Displacement "

FIG. 83.—Comparison of valve displacement—Walschaerts and Stephenson valve motions.

rapid motion to the valve due to the proportions of the lap and lead lever. This is an interesting feature of the gear, and can be seen when following it through its motions.

The *total* valve displacement with each gear is necessarily the

same, as each gear was designed to give a desired maximum valve travel. But for any particular crank angle, or per cent. of stroke, it will be noted that the two gears do not give the same displacement of the valve.

For instance, to find the valve displacement at 40 per cent. stroke, and the reverse lever in, say, the fifth notch: Follow across from the 40 per cent. line to the piston displacement curve; then follow the ordinate until it cuts the valve displacements curves, the heights of which from the center line will give the valve displacement from its mid-position, for that respective position of the link block. It will be noted that they are different for each gear.

QUESTIONS ON CHAPTER X

1. Without referring to an illustration, make a careful free-hand sketch of a locomotive boiler, naming 15 leading parts.

2. What would be the advantage and disadvantage of a feed-water pump instead of an injector for a locomotive?

3. Show that a boiler horse power also equals the evaporation of 30 lb. of water at a temperature of 100°F. into steam at 70 lb. pressure above atmosphere.

4. Why is a square foot of superheating heating surface less effective for absorbing heat than is a square foot of saturated steam surface? What factor in Eq. 82 would be different in the two cases?

5. In what way is the water rate of a locomotive engine found to vary with (*a*) speed; (*b*) initial pressure; (*c*) back pressure? Give a summary of the results by Goss on initial pressure conditions.

6. Name the chief advantage noted for the Stephenson gear; the advantages of the Walschaerts gear.

7. State carefully, step by step, the development of the Walschaerts valve motion. What is meant by "constant lead" and how is it obtained in Walschaerts valve gear?

8. Explain by the Zeuner diagram, Fig. 82, why the lead is constant for all positions of the reverse lever.

9. Lay out a Walschaerts valve gear for inside admission, taking the leading dimensions either from an engine on the track or from drawings supplied, following the suggestions for design and drawing the Zeuner or the Bilgram diagram for the valve motion. (The solution of this will require information outside of this chapter.)

10. From Fig. 83, point out the advantages over the Walschaerts gear of the valve motion derived from the Stephenson gear when the valve travel is 4⅞ in.

CHAPTER XI

SUPERHEATED STEAM: THEORY AND PRACTICE

TYPES OF SUPERHEATERS; DESCRIPTION OF SCHMIDT SUPERHEATER; THEORY AND ADVANTAGES OF SUPERHEAT; ENTROPY DIAGRAMS APPLIED TO SUPERHEATED STEAM; CYLINDER CONDENSATION AVOIDED BY USING SUPERHEATED STEAM; QUALITY OF STEAM AT ANY POINT; WATER RATE FROM THE ENTROPY DIAGRAM; VARIOUS COMPARISONS OF TESTS OF SUPERHEATED STEAM LOCOMOTIVES

Introductory.—Robert Trevithick, one of the very first to develop the steam locomotive, was also a pioneer in the practical application of superheated steam. His superheater was tried out on a condensing pumping engine in 1828 and gave a saving in coal of no less than 33 per cent. This design embodied the features of the present day fire-tube superheater, the only type which has come into wide use in locomotive practice. The carefully conducted experiments of Hirn in 1857, showed that on a simple stationary engine with boiler pressure of 55 lb. and 100° to 190°F., superheat, a steam saving of 20 to 47 per cent. could be obtained. The development of the locomotive superheater since then is the result of the efforts of many engineers. The following four types, applied with varying success, have been developed:

1. The fire-tube type, in which separate superheater elements are placed inside a special flue, which is from 5 to 5½ in. in diameter.

2. The smoke-box type, the superheater (in whole or in part) being placed in the smoke box.

3. The boiler-barrel type, in which part of the heating surface of the fire-tubes of the boiler is used as superheating surface.

4. The fire-box and flue type in which a part or all of the superheater heating surface is placed in the fire-box.

It is pointed out by Mr. C. D. Young in a review of locomotive superheaters and their performance, that the fire-tube type now in general use, is essentially the same as one invented by De

Montchenil in France about 1850 and which has been developed and applied within the past 15 years under patents of Dr. Wilhelm Schmidt. Schmidt also developed other types, notably a fire-box superheater which was used on the continent for some years.

The Schmidt Superheater.—The prevailing type of locomotive superheater is shown in Fig. 84. Each superheater element is made up of four seamless steel tubes, approximately 1½ in. in diameter. They are so constructed by cast steel return bends that steam from the boiler after passing through a header, *A*, in the smoke-box, flows into one of the superheater elements,

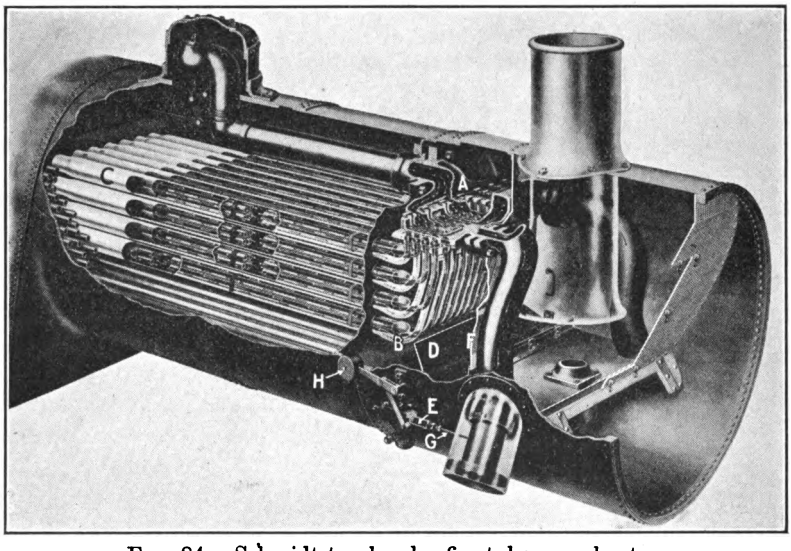


FIG. 84.—Schmidt top-header fire-tube superheater.

through which it passes to the fire-box end, thence to the front end, and back again to the fire-box and returning to the superheater header as superheated steam of approximately 650°F., thence through the steam pipes in the front end to the valve chests and cylinders. The two headers are in one casting, as shown in the illustration.

The flow of gases through the large fire-tubes (or flues) *C* is controlled by a damper *D*, which is operated by the damper cylinder *E*, this being located on the outside of the smoke-box on the engineman's side of the locomotive. The normal position of the damper is closed when the throttle of the locomotive is

similarly closed. When, however, the throttle of the locomotive is opened, thus permitting steam to pass into the steam chest, pressure accumulates through connecting pipe *G* to cylinder and piston *E*, which pressure raises by means of bell cranks, a counterweight and opens damper *D*. The fire-box gases are then free to flow through the large tube *C*, past the superheater element *B*, giving up part of their heat to the steam passing through the superheater element. As soon as the throttle is closed, piston in cylinder *E* is relieved of pressure and counterweight *H* falls of its own weight, closing *D*, thereby protecting the superheater elements *B* from the flow of hot gases from the fire-box. The engineman should observe the position of counterweight *H* in order to determine whether the damper is closed when steam is shut off and if open when steam is being used; for if the counterweight is down and damper closed while running, little, if any, superheat will be formed.

Each group of superheater coils, located inside the flue *C*, extends to within a foot or two of the back flue sheet and is held in the flue by supports, arranged to obstruct but little the flow of gases.

Theory Concerning Superheat.—If heat is added to steam while in contact with water, more water will be evaporated and the quantity of steam will thus be increased, but the temperature remains the same if the pressure is kept constant. If heat is taken from it, either by cooling or by performing useful work during expansion, a part of the steam is condensed and unless it is re-evaporated by the absorption of more heat, it loses its capacity to do work. This is a serious loss, for cylinder condensation at the point of cut-off may amount to 25 per cent. or even more.

If steam, after being evaporated, is still further heated, its temperature will be raised above that corresponding to saturation and it becomes superheated. Steam thus highly heated may be cooled through the range of superheat before cylinder condensation takes place and since superheated steam is a poor conductor of heat, the actual heat absorbed by the superheated steam is less than it is for the same weight of saturated steam. *The prevention of cylinder condensation is one of the notable advantages of superheat.*

A second advantage results from its increased volume per unit weight over that of saturated steam. *The higher the temperature*

to which steam is superheated at a given pressure, the larger is the volume of steam which can be produced from each pound of water. Based on this property, many engineers favor a high degree of superheat. If a saturated steam locomotive is working at 200 lb. (gage) pressure, the temperature of the entering steam will be 387.9°F., the specific volume 2.141 cu. ft. (represented as V, Fig. 85) and the total heat per pound 1199.2 B.t.u. Suppose 200°F. superheat be applied to this locomotive, without changing the size of the cylinder or the speed, then the specific volume will be 2.84 cu. ft. (represented as V_1, Fig. 85) which is 32.7 per cent. greater than the volume of saturated steam; the total heat per pound will be 1309.7 B.t.u., an increase of 9.2 per cent. over saturated steam. *The gain is* 32.7 *per cent. in volume by adding but* 9.2 *per cent. to the total heat.* Since the cylinder takes in the same volume for each stroke at the same cut-off for both dry saturated and superheated steam, the weight of superheated steam used per stroke for the same work will be in round numbers 30 per cent. less. Dr. Schmidt estimates that the total economy in water obtained by superheated steam of 170 lb. and 650°F. will be approximately:

10 per cent. on account of increased volume of steam; and

30 per cent. on account of avoiding losses due to cylinder condensation.

In actual service, the water economy varies from 15 to 50 per cent. or more, according to the cut-off, speed and to other conditions or running. The best results are obtained on long continuous runs where uniformly high superheat is maintained and where worked under nearly constant conditions. In general, the harder a superheater locomotive is forced, the higher is the degree of superheat obtained and the more economically the engine works; whereas, the opposite is usually the case in the saturated steam engine, one cause of the loss being from water carried over into the cylinder when the boiler is being forced.

Cylinder Action.—The saturated steam entering the cylinder comes into contact with and gives up heat to the metal of the cylinder and to the steam passages and piston, which have been cooled during the preceding period of expansion and exhaust. A considerable part of the entering steam is condensed before cut-off and is deposited over the surface in a finely divided state. This condensed steam has lost its capacity for external work. The condensation continues during expansion up to a

point where the temperature of the expanding steam reaches the temperature of the cylinder walls. From this point to the end of expansion, and especially during exhaust, the heat previously absorbed by the walls is returned and some of the water in the steam is re-evaporated. *Re-evaporation has been found to increase with the wetness of the boiler steam and to decrease with the longer cut-offs and higher speeds, a condition which checks with theory.* Since the re-evaporation occurs (for the most part) after release, the heat thus taken up is useless, for it is carried away by the exhaust steam; in fact, it may increase the back-pressure.

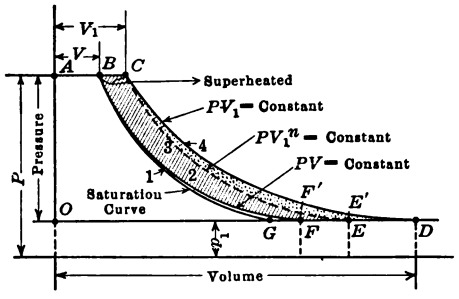
Superheated steam absorbs heat during admission, causing a reduction of temperature of the gas in contact with the walls and an increase in temperature during exhaust. Since the heat to raise a unit weight of superheated steam through the necessary temperature is small compared with the heat absorbed in vaporizing the same weight of water (which has been condensed and distributed over the surface of the cylinder), it follows that the heat taken from the cylinder walls is relatively small in the case of superheated steam. For example, it requires about 75 B.t.u. to raise the temperature of a pound of superheated steam through 150°F. but it requires eleven times that amount of heat to vaporize 1 lb. of water into steam at 200 lb. pressure.

Since superheated steam has a temperature higher than that corresponding to the point of condensation and can give up the heat in the superheat without condensing any of the steam, it follows that steam may be superheated sufficiently to entirely avoid cylinder condensation.

In general, steam should leave the locomotive cylinder with but very little, if any, moisture and this requires "high" superheat at admission. Further, heat exchanged during admission becomes less as the amount of superheat increases.

Expansion of Superheated Steam.—Figure 85 presents a theoretical pressure-volume diagram for 1 lb. steam (*a*) saturated, shown by the expansion lines 1 and 2 and (*b*) superheated, by lines 3 and 4. Curve 4 represents the ideal expansion of superheated steam following the same thermodynamic law as the (theoretical) saturated steam curve. But superheated steam in a cylinder follows a different law during expansion, being represented by *CE* or curve 3. Note that the line *CE* falls to the left of the line *CD*, the latter being the line of isothermal expansion

or PV_1 = constant. But superheated steam, expanding by the law PV^n = constant, does less work than is done by saturated steam expanding from the same cut-off, by an amount equivalent to the area *C–D–E–C*, or by *C–E′–E–C* if the volume at the end



$AB = V$ = Volume 1 lb. saturated steam.
$AC = V_1$ = Volume 1 lb. superheated steam.
Curve 1 = Saturated steam line.
Curve 2 = Saturated steam, isothermal expansion.
Curve 3 = Superheated steam, adiabatic expansion.
Curve 4 = Superheated steam, isothermal expansion.

FIG. 85.—Expansion lines for saturated and for superheated steam.

of expansion is the same in both cases. At 25 per cent. cut-off, the area *C–D–E–C* is about 7 per cent. of the area *A–C–E–O–A*.

By principles of geometry, the higher the value of the exponent "*n*" of the curve, the steeper will be the line of expansion of the superheated steam curve PV^n = constant; and since the value of this exponent increases with the degrees of superheat, it

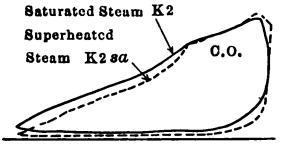


FIG. 86.—Cards for equal cut-off.

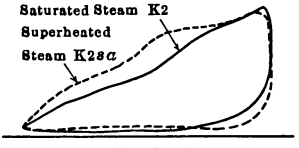


FIG. 87.—Cards for equal weight of steam.

Cylinders..... 24 × 26 in.
Boiler pressure..... 205 lb.

Speed 180 r.p.m..... 42 m.p.h.	K2sa	K2
Superheat, degrees.....	236	None.
Cut-off, per cent.....	37	37
M.E.P., lb.....	88.0	84.8
I.h.p.....	1,849.4	1,816.0
Steam per hour, lb.....	33,421	48,553
Steam per i.h.p. hour, lb..	18.1	26.7

Speed 160 r.p.m..... 38 m.p.h.	K2sa	K2
Superheat, degrees.....	268	None.
Cut-off, per cent.....	51	37
M.E.P., lb.....	110.2	88.7
I.h.p.....	2,047.8	1,657.1
Steam per hour, lb.....	39,049	40,558
Steam per i.h.p. hour, lb..	19.1	24.5

follows that the difference between the areas under curves 3 and 4 increases as the superheat is increased. This is of less practical than it is of theoretical interest, for the advantages of superheat mentioned in this chapter far outweigh its disadvantage here noted.

In Fig. 86 is brought out an interesting confirmation of the

theory concerning expansion. With the same cut-off, the saturated steam card continues the expansion *above* that of the superheated steam, as in the theoretical diagram, Fig. 85. The mean effective pressure in Fig. 86 is found to be greater for the superheated steam card but this is due to less back-pressure. If the back-pressure was the same in both cases, the M.E.P. and, therefore, the work done by the saturated steam engine, would be the greater of the two; but for all practical purposes they may be considered the same under the conditions here stated. In the case of saturated steam, re-evaporation occurs during exhaust and this may, in part, account for the greater back-pressure in the saturated steam card.

A better basis for comparing cylinder performance of saturated and superheated steam is shown in Fig. 87, also taken from tests at Altoona. Each of these cards is for nearly an equal weight of steam at a speed of 38 m.p.h. The diagram of the K2sa gives a higher M.E.P. and consequently a greater horse power, amounting in this case to 28 per cent. favorable to the superheated locomotive.

Indicator Card for Superheated Steam.—The indicator card *C–L–H–I–J*, Fig. 88, is from a Pacific type locomotive at a speed of 180 r.p.m. (42.8 m.p.h.), 31 per cent. cut-off and with 236° superheat at the steam chest. The card shows a drop of 16 lb. from boiler to initial cylinder pressure and a further drop of 52 lb. to cut-off pressure. This pressure drop up to cut-off, due to wire-drawing the steam, causes a serious loss in output of the engine, being represented by *C–L–E–C*. Had the pressure for 31 per cent. cut-off been at *E*, the expansion would follow the adiabatic line *EF* and the shaded area *C–E–F–H–C* represents the total additional work which would theoretically be available without any pressure drop up to cut-off. It should be noted that this possible increase has to do with conditions for steam entering and while in the cylinder, rather than to the behavior of the steam itself.

The line *DG* has been drawn to represent the condition without superheat, but with the same weight of steam for each stroke; that is, the same weight of saturated steam at the same initial pressure would be cut off at *D*, instead of at *E* (for saturated steam) and would expand along *DG*, the theoretical gain by this superheating being represented by the area *D–E–F–G–D*, expressed in foot-pound units. In the card here shown, this gain

e, *e–f–g* will be a vertical line, that is, a line of constant entropy, and the area *d–e–f–d*, to the right of the saturation curve, is the field or area of superheat. From *f* to *g*, the steam is wet, the percentage of moisture increasing as the temperature falls to back-pressure conditions.

For adiabatic expansion, the efficiency is:

$$\text{for saturated steam, } E = \frac{\text{work done} = b\text{–}c\text{–}d\text{–}h\text{–}b}{\text{heat absorbed} = k\text{–}b\text{–}c\text{–}d\text{–}j\text{–}k}$$

$$\text{for superheated steam } E' = \frac{b\text{–}c\text{–}d\text{–}e\text{–}g\text{–}b}{k\text{–}b\text{–}c\text{–}d\text{–}e\text{–}i\text{–}k}$$

The gain in heat units due to superheating above the temperature corresponding to that at which it will vaporize at a given pressure is represented by area *d–e–g–h–d*. Superheated steam is found by the above to be more efficient than saturated steam *of the same pressure;* but it is not as efficient as saturated steam *of the same temperature.*

We may repeat, that if there is no loss in temperature from boiler to the point, *e*, of adiabatic expansion, *b–c–d–e* becomes the compression and admission lines of the theoretical indicator diagram.

The Corrected Heat Diagram.—Figure 88*a* is the HEAT OR ENTROPY DIAGRAM of the indicator card shown in Fig. 88. Areas in this diagram represent a definite amount of heat. The scale of heat units is shown to the right, the square representing 2.5 B.t.u. After the steam has all been vaporized in the boiler, it is superheated and enters the steam chest at the observed temperature of 617°F. or 236° superheat, assuming no drop in temperature from steam chest to cylinder. From the point *R* where the steam is superheated and contains 1325 B.t.u., it expands adiabatically or nearly so, along the line *RZ*, the temperature dropping and release occurring at *S*, while the steam still contains some superheat.

Let *OMWPQX* represent the heat contained in 1 lb. of saturated steam at pressure and temperature *Q*, and let *XQRZ* represent the heat added to superheat. Then, if the superheated steam in the cylinder be expanded down to back pressure *WU*, the steam at release would be dry saturated steam without any superheat, and the efficiency of the superheat = *V–Q–R–U–V* ÷ *X–Q–R–Z–X*. For the case where steam is superheated at release, if the steam in the cylinder at some high

temperature, R, is expanded along the adiabatic line R–Z to some release pressure as S, where it is still superheated, the superheated steam will then follow the constant-volume lines $STTH'$ until it falls to the back-pressure line WH'. The efficiency of the superheat is T'–Q–R–S–T–T' ÷ X–Q–R–Z–X, and the loss due to release taking place before all of its superheat had been used is V–T'–T–S–U–V. The heat-equivalent of these areas can be measured from the temperature-entropy chart in heat units.

The constant volume curve from T is drawn by taking the specific heat of steam at constant volume and drawing the curve TS as QR was drawn for constant pressure or it may be plotted from values taken from the steam tables.

This diagram brings out a fact not usually recognized, that no important theoretical gain on solely a heat-unit basis can be realized from superheating.

Without the refinement requiring a complete thermodynamic analysis, the original indicator card shown in Fig. 88 would have the form I'–J'–C'–Q–R–S–T–H'–I' (Fig. 88*a*), and without superheat, the card would be I'–J'–L'–T'–H'–I', an area considerably less than for superheat. This study gives a definite conception of actual heat losses in the cylinder.

Problem 26.—Construct on a cross-section sheet the diagram for the theoretical entropy card at 180 lb. initial gage pressure and 20 lb. back pressure (*a*) for saturated steam, (*b*) for 100, for 200 and for 300°F. superheat, and find the per cent. gain in work for each of the three superheats over saturated steam. Take into account incomplete expansion, but not the effect of steam in clearance.

To obtain the quality at the point V, if the expansion were adiabatic from the point Q, the following equation may be solved for X_2

$$\frac{X_2 L_2}{T_2} + N_2 = \frac{L_1}{T_1} + N_1 + C_p \log_e \frac{T}{T_1} \qquad (97)$$

where in general the subscript 1 represents the higher pressure and the subscript 2 the lower pressure condition, T being the absolute temperature of the superheat and C_p the specific heat of superheated steam at constant pressure. The separate terms in Eq. 97 are represented (in their respective order) in Fig. 88*a* by $WU + JW = I^1V + JI^1 + VU$. We may express the quality of the steam at V by the ratio WV ÷ WU, which in this case is 88 per cent.

Our study may be extended to determine the approximate water rate of the engine. As stated, each pound of steam at *R* contains 1325 B.t.u. In the exhaust as represented at point *U*, it has a total heat per pound 1165 B.t.u. (corresponding to saturated steam at 255°F.). The difference between the above is 160 B.t.u. The equivalent work done per pound of steam, if all the heat could be used is, therefore, $160 \times 778 = 124{,}480$ ft.-lb. This work can be expressed as a water rate; for since each horse power is equivalent to 1,980,000 ft.-lb. per hour, $1{,}980{,}000 \div 124{,}800 = 15.9$ lb. of water per horse power, would be the least possible water rate which could be secured with an exhaust temperature of 255°F., corresponding to 32.5 lb. absolute back-pressure. The test during which the card in Fig. 88 was taken gave 16.4 lb. but the observed exhaust temperature in the exhaust passages was more than 255°F., in fact it showed 84° of superheat, bringing the actual water rate calculated as above, even more nearly to the theoretical. This method forms a valuable check on tests, for the water rate as calculated above (based on the heat content for initial and final temperature in the cylinder) cannot be *below* that obtained by weighing the water fed to the boiler.

Compounding Superheater Locomotives.—In discussing Fig. 70, Chap. X, attention was called to the advantage of high-pressure steam, the augment of which required but a small additional percentage of total heat. When the steam is expanded in two stages, this advantage is even more apparent. Suppose the ratio of low-pressure to high-pressure cylinder volume is 1:2.5 and cut-off is at 45 per cent. in the high-pressure cylinder; the nominal expansion will be $2.5:0.45 = 5.55$, while in a single expansion engine it would be but $1:0.45 = 2.22$. This advantage is of little account at high speed and early cut-off, the principal reason for this being the marked drop in cut-off pressures at high speed, as shown by cards in Fig. 68.

Attention has also been called to the loss caused by initial condensation, resulting from steam at the temperature corresponding to initial pressure coming into contact with the cylinder walls which are at a lower temperature, due to the exhaust of the previous stroke. This difference of temperature is less in a compound engine, because the drop in pressure and hence, the drop in temperature, is less. However, this advantage is partly offset by the larger surface exposed to the steam in the compound

cylinders. Taking into account all theoretical and practical conditions, including maintenance, compounding saturated steam in locomotives is of questionable net saving at the higher speeds.

For reasons, carefully stated in this chapter, superheated steam offers a distinct advantage in that less heat is taken up by the entering steam and initial condensation is eliminated.

There are three methods of superheating in compound locomotives: (1) initial superheating; (2) two-stage or intermediate superheating; and (3) low-pressure cylinder superheating, sometimes called intermediate superheating.

In the third method, steam is superheated only after leaving the high-pressure cylinder. This does not better the conditions in the high-pressure cylinder.

The problem reduces to one of relative advantage of methods (1) and (2) which may be stated as follows: Assuming dry steam at low-pressure exhaust, is it an advantage to supply initial superheat rather than apply two superheaters, one of which is between the high- and low-pressure cylinders? In 1901 tests were run on a tandem compound, 2–8–0 type, of the A. T. & S. F. Ry., equipped with high- and low-pressure superheaters. Both superheaters were in the front end, the low-pressure superheater being directly ahead of the front flue sheet and the high pressure in front of the low-pressure superheater. The data taken on the tests were sufficiently complete to enable one to study the question, at least theoretically, of the relative advantage of using one superheater giving initial high superheat as against the same total heat in superheating, divided into a "high" and a "low" pressure superheater.

The author analyzed this problem and reached the following conclusion:[1] (1) That a small amount of superheating in the high-pressure cylinder is of little or no advantage; (2) when the pressure drop is large between the cylinders, large gain is effected by a considerable superheat in the low-pressure cylinder; (3) for the same amount of heat in the superheat a less amount of heat could be (theoretically) transformed into work, by using the two superheaters, than could be obtained by the same amount of heat from one high-pressure superheater.

The results of the study of the Santa Fe tests favorable to method (1) are confirmed by theory. On the other hand,

[1] The Stevens Institute Indicator, Vol. XXVIII, No. 1.

the net gain would not *theoretically* be as great by initial superheating as by introducing on a locomotive four single expansion cylinders with superheat to prevent condensation during expansion.

The pressure-volume diagram, Fig. 91, shows an actual card, nearly perfect, and approaching that of a Rankine cycle diagram. The expansion line for steam superheated initially 250°F. and expanded adiabatically has been added. It will be noted that the superheated steam curve of expansion crosses the saturation curve (shown dotted) near the end of expansion of the high-pressure cylinder. This indicates that sufficient superheat was applied to avoid condensation in the high-pressure cylinder.

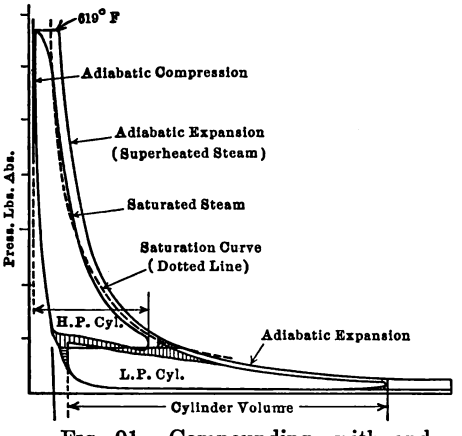


FIG. 91.—Compounding—with and without superheat.

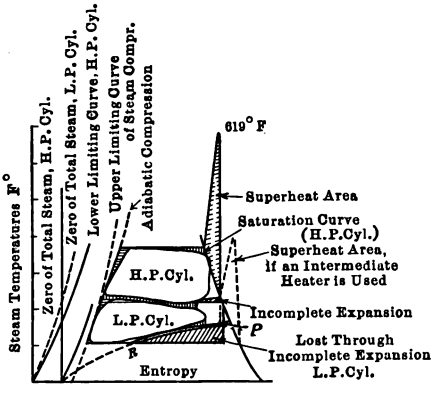


FIG. 92.—Entropy diagram of Fig. 91.

The conditions are brought out more pointedly in Fig. 92 in which the several losses from the ideal card are indicated and the gain by superheating in the high-pressure cylinder is shown by the area to the right of the saturation curve. This diagram shows that with superheat sufficient to prevent condensation in the high- but not in the low-pressure cylinder, the release follows *P–R* and the loss from incomplete expansion in the low-pressure cylinder would be but little more than in the saturated steam card. If intermediate heating were used, the work area added to the low-pressure cord is shown dotted in Fig. 92.

The problem, therefore, is one of determining the sum of the losses and the gains for any particular problem, taking into account

all the conditions. On the basis of the above suggestions, the student with a working knowledge of the theory of heat engines, may analyze completely this vital problem.

Tests of Superheater Locomotives.—The results of tests taken from both testing plant and road trials confirm the advantages which theory indicates for economy of superheating. There are many methods of comparing results, three of which may

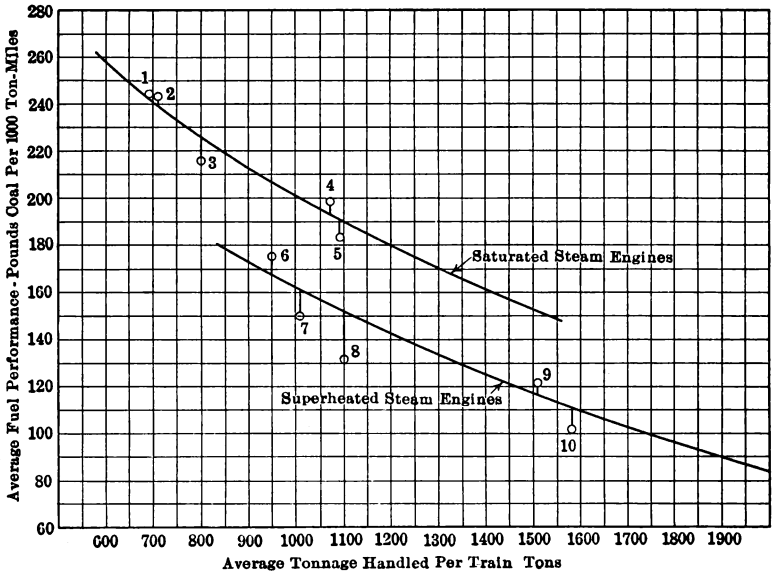


FIG. 93.—Comparative performance of freight locomotives on Seaboard Air Line.—Ruling grades 1–2 per cent.

Saturated
1. Ten-wheel Locomotives, Virginia Division.
2. Ten-wheel Locomotives, North Carolina Division.
3. Ten-wheel Locomotives, Georgia Division.
4. Consolidation Locomotives, North Carolina Division.
5. Consolidation Locomotives, Virginia Division.

Superheated
6. Pacific Type Locomotives, North Carolina Division.
7. Pacific Type Locomotives, Georgia Division.
8. Pacific Type Locomotives, Virginia Division.
9. Mikado Type Locomotives, North Carolina Division.
10. Mikado Type Locomotives, Virginia Division.

be noted: A. to select a saturated and superheater locomotive of the same general dimensions, hauling trains over the same division under regular traffic regulations, comparing the maximum tonnage hauled by each; B. special test runs over the same division by locomotives especially selected for the trials, comparing coal and water consumption; C. locomotive testing plant results, in which locomotives of the same type and general

dimension, saturated and superheater, are each mounted on wheels and compared in much the same way as engineers study results from stationary power plants. Results from each method will be given.

A. Figure 93 shows comparative results from performance reports for October, 1914, obtained from working a number of superheated and saturated steam locomotives on division of the Seaboard Air Line having the same maximum grades.[1] Locomotives of the Ten-wheel, Consolidation, Pacific and Mikado types were taken for the comparison. The Mikados had been in service only a few weeks and those of the Pacific type a little over a year; both were equipped with superheaters and brick arches.

The height from the base line represents the fuel consumption per ton-mile, while the distance to the right represents the average tonnage per train hauled by each group of engines. For instance, the average fuel consumption of the ten-wheel engines, Virginia division (point 1) is 244 lb. per 1000 ton-miles, and the average weight of train hauled is 690 tons. The Mikado type engines on the same division (point 10) averaged 102 lb. of coal per 1000 ton-miles and 1580 tons hauled per train.

The points representing the performance of each group of saturated steam locomotives lie approximately upon one curve, while the points representing the performance of the superheater locomotive lie upon another curve somewhat lower than the first. The distance between these two curves represents approximately the saving in fuel per ton-mile effected through the use of superheated steam on engines of various sizes and types.

Some preference in the character of trains hauled may have been given the Mikados, but the greater capacity and economy of this type in comparison with the other locomotives is remarkable. In this instance the Pacific type engines appear to be rather an extravagant type for freight service, as they not only burn considerably more coal per ton-mile than the Mikados, but their average speed is lower. Both of these types are operated under the same maximum speed limit, hence the Pacific type is at a disadvantage on ascending grades and in accelerating trains.

B. **Norfolk and Western Road Tests.**[2]—These tests were conducted on freight locomotives of the following dimensions:

[1] Reported by L. G. Plaut, Fuel Engineer, in Railway Age Gazette, March 5, 1915.

[2] Reported by H. W. Coddington, Engineer of Tests, Norfolk and Western Railway, A. R. M. M. A., 1914.

TABLE XVIII

		Eng. No. 1160[1] (superheated)	Eng. No. 1136 (non-superheated)
1	Type of locomotive	4-8-0	4-8-0
2	Total weight (3 gages of water)	262,000	263,000
3	Total weight stoker	.....	5,220
4	Total weight on drivers (Workg. Ord.)	222,000	228,000
5	Cylinders (normal)	24 by 30 in.	24 by 30 in.
6	Diameter of drivers (actual)	55½ in.	54¼ in.
7	Heating surface (fire-box)	179 sq. ft.	179 sq. ft.
8	Heating surface (arch tubes)	.....	13
9	Heating surface (superheater flues)	690	.....
10	Heating surface in tubes (small)	2,103	4,281
11	Heating surface in tubes (large superheater)	905	.....
12	Heating surface in fire-box (inc. arch tubes)	192	192
13	Total heating surface (without arch)	3,877	4,473
14	Total heating surface (with arch)	3,890	.....
15	Grate area	45	45
16	Boiler pressure (normal)	200	200
17	Valves—type	15 in. pis.	15 in. pis.
18	Valves—motion	Baker	Walsch't
19	Fire-box—type	Wide	Wide
20	No. of tubes (small)	217	386
21	No. of flues for superheater	34	.....
22	Outside diameter of tubes	2	2¼
23	Outside diam. of flues	5½	.....
24	Length of tubes	18 ft. 6 in.	18 ft. 10 in.

[1] This locomotive, equipped for test, is shown in Fig. 103, page 226.

The results have been summarized as in Table XIX.

Only two of the items are observed to show favorably to the saturated steam locomotive, items 6 and 7.

The average boiler efficiency for the superheated locomotive exclusive of the superheater is 47.9 per cent. The proportion of the total boiler efficiency credited to the superheater is 4.6 per cent. Item 7 gives the total boiler efficiency, including the superheater, or 52.5 per cent., which is slightly lower than the 59.9 per cent., the average boiler efficiency of the non-superheater locomotive. This difference in boiler efficiency of 12.3 per cent. favorable to the non-superheated locomotive is more than offset by the economical influence of the superheated steam upon the locomotive as a unit. An average boiler horse power of 1103.8 is attributed to the boiler exclusive of superheater. The proportion of the boiler performance which may be attributed to the superheater alone amounts to 108.3 b.hp. The

total is 1212 b.hp., *and is almost identical with the results obtained from the non-superheated locomotive.*

TABLE XIX.—SUMMATION OF PERFORMANCE OF SUPERHEATED AND NON-SUPERHEATED LOCOMOTIVES, NORFOLK AND WESTERN RAILWAY

Item	Eng. 1160 superheated	Eng. 1136 non-superheated	Per cent.[1] difference
1 Boiler pressure	199.2	192.7	+ 3.3
2 Tonnage hauled	1,198.9	1,032.0	+16.1
3 Speed m.p.h.	19.23	14.78	+30.1
4 Coal per M. ton-mile	244.3	334.0	+26.8
5 Water supplied to boiler	49,622.0	74,551.0	+33.4
6 Equivalent evaporation per lb. of dry coal	7.51	8.32	− 9.7
7 Boiler efficiency	52.5	59.9	−12.3
8 Steam through cylinders	45,257.0	68,356.0	+33.8
9 Steam through cylinders per M.t. car mile	1,270.0	2,184.0	+41.8
10 B.t.u. through cylinders per M.t. ml.	1,410,213.0	2,188,011.0	+35.5
11 Indicated hp. average for run	1,609.9	1,182.8	+36.1
12 Drawbar hp. average for run	1,410.2	1,004.5	+40.4
13 Steam per i.hp. per hour—actual	20.0	28.2	+29.1
14 Steam per i.hp. per hour—equiv.	25.9	35.5	+27.0
15 Mechanical efficiency	87.58	84.98	+ 3.0
16 Thermal efficiency i.hp. basis	5.31	4.29	+23.8
17 Thermal efficiency d.b.hp. basis	4.65	3.65	+27.4
18 Fuel burned per hour	5,635.00	5,097.0	+10.5
19 Fuel burned per sq. ft. grate area per hour	125.2	113.2	+10.6
20 Fuel burned per hour—high speed condition	5,243.0	4,699.0	+11.6
21 Fuel burned per hour—low speed condition	5,941.0	5,347.0	+11.1
22 Fuel burned per sq. ft. of grate area per hour—low speed	132.0	119.0	+10.9
23 Fuel burned per sq. ft. of grate area per hour—high speed	116.5	104.4	+11.6

[1] All differences preceded by plus (+) are favorable to the superheated locomotive.

Figures 94 and 95 suggest a further study under method B, the results being taken from road tests with a K2 and K2s engine of the same dimensions. Figure 95 shows the characteristics of the superheater locomotive giving the best results at the higher speeds and gives results of an acceleration test with eight steel

cars. Note that after the second minute the d.hp. of the superheater locomotive exceeded the saturated locomotive, there being less falling off of the d.hp. of the superheater locomotive than the saturated locomotive as the time from starting increased. (Note results in Fig. 12, Chap. III.) These locomotives are able to take approximately a 500-ton passenger train from a standing start to 80 m.p.h. in from 10 to 12 minutes, or an acceleration to this speed at the average rate of 0.13 m.p.h. per second for the superheater locomotive, and 0.11 for the saturated locomotive.

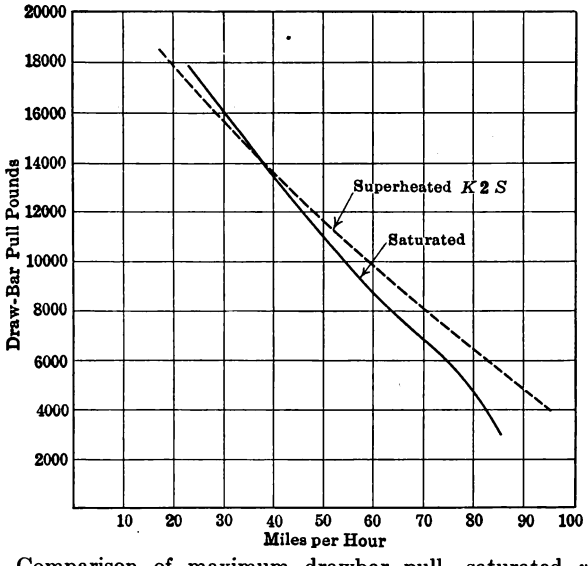


FIG. 94.—Comparison of maximum drawbar pull—saturated *vs.* superheated steam locomotives.

There was an indication of a 5 per cent. saving in cost per M. ton-miles in the performance of the superheater locomotive with a brick arch as compared with the same locomotive without the arch.

The superheated locomotive handled 16.1 *per cent. more tonnage at an increase of* 30.1 *per cent. higher speed and at the same time showing an economy of* 26.8 *per cent. in coal per M. ton-miles.*

C. *Locomotive Test-plant Results.*—The elaborate tests on the Purdue University testing plant, reported by Deans, Goss and Benjamin and Professor Endsley, and on the Pennsylvania

Railroad plant at St. Louis and later at Altoona, reported by Mr. E. D. Nelson, formerly Engineer of Tests, and by Mr. C. D. Young, now Engineer of Tests, form the most complete records of locomotive performances ever presented. In addition to locomotive testing-plant results previously referred to in former chapters, the following may be noted.

The frontispiece shows a K2sa in position for test at the Altoona plant. As a result of extensive tests on their locomotive, the following general conclusions were reached:

1. As the superheat is reduced, the evaporation of the boiler

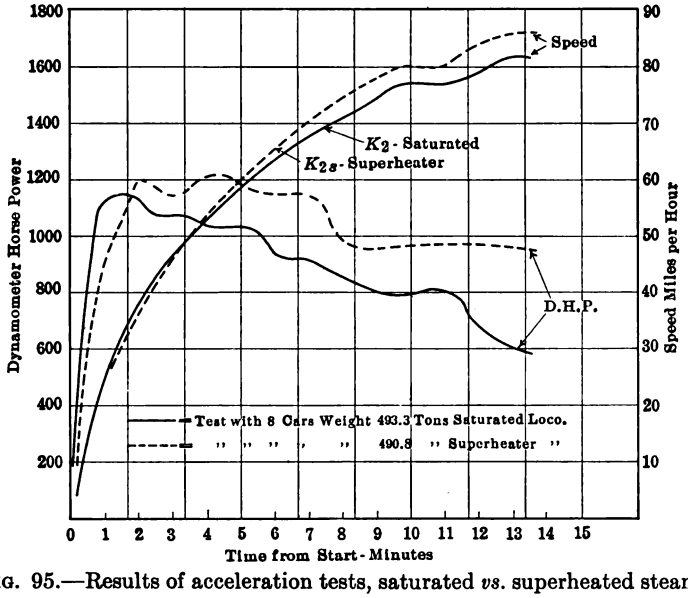


FIG. 95.—Results of acceleration tests, saturated *vs.* superheated steam.

is increased within certain limits; in other words, a boiler without superheater shows a larger maximum evaporation than one with a superheater. The power of the locomotive, however, does not increase with the greater weight of steam produced; on the contrary, the power is reduced with the reduction in superheat.

2. Within the limitations of the tests, the highest superheat does not result in the lowest water rate; this is on account of the fact that to obtain the highest superheat the locomotive may be run at an excessively long cut-off, the long cut-off increasing the water rate to a greater extent than is compensated for by the increase in superheat.

3. It is seen that the advantage of superheating may be utilized in two ways; either in coal and water saved, due to a reduced water rate, or by burning the same amount of coal as would be required in the boiler where it is generating saturated steam and obtaining a decided increase in the power output of the locomotive. If we exclude conditions of starting, this would permit superheater locomotives to haul heavier trains with a saving in transportation facilities and labor.

In order to obtain different degrees of superheat without in any way changing the water-heating surface of the boiler or the engine conditions, different forms of superheater elements

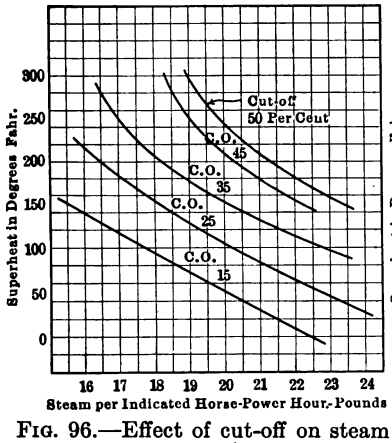


FIG. 96.—Effect of cut-off on steam consumption.

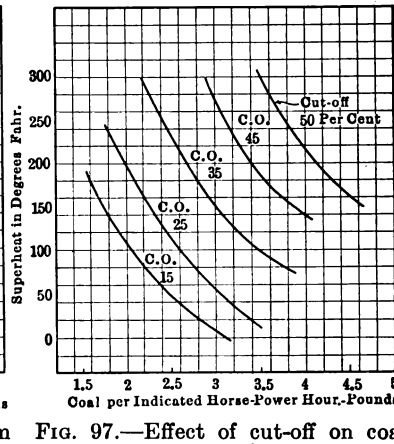


FIG. 97.—Effect of cut-off on coal consumption.

were used, maintaining the same superheater header and large flues.

Figures 96 and 97 express graphically some results of these experiments. Figure 96 shows that for every increase in superheat at any cut-off, there is a saving in steam; and that at 15 per cent. cut-off, for every 20° rise in superheat, there is a reduction in water rate of 1 lb. per indicated horse power hour.

With a cut-off of 15 per cent. and a superheat of about 70° we can obtain a water rate of 19 lb. per indicated horse-power hour, while if the cut-off is extended to 50 per cent. at the same speed, the superheat must be increased to 300° so that the water rate will remain at 19 lb. The very great importance of the length of cut-off is thus apparent, but even with the longest cut-off that is shown the water rate does not reach 24 lb., or what

may be called the minimum water rate for saturated steam. It thus appears that superheat always shows a saving in steam if the cut-off does not exceed 50 per cent. Very similar conclusions may be drawn from Fig. 97, which shows the superheat, the cut-off and coal per horse-power hour.

A Testing-plant Product.—The latest E6s Atlantic, or 4–4–2 type locomotive of the Pennsylvania Railroad, has been designated by London *Engineering* as "A Locomotive Testing-plant Product."[1] This engine has a maximum output about 15 per cent. greater than the earlier design with shorter tubes, while the efficiency is also higher. Another important change is in the size of the cylinders. The earlier engine had cylinders 22 by 26 in.; the new engine has cylinders 23.5 by 26 in. This increase was based on results already noted, that with superheated steam, the higher maximum power obtained was developed with a much later cut-off than with the saturated steam engine. At 75 m.p.h., with 35 per cent. cut-off, this engine develops power equal to that produced by the earlier type at 40 per cent. cut-off. This later engine, however, seems to develop its maximum power at a somewhat longer cut-off and slower speed, whereas the short-boilered engine produced its greatest power at its highest speeds. The piston valve diameter for the later engine was reduced from the standard 14 in. to 12 in. with a reduction in weight from 218 to 120 lb., and a consequent reduction in the value stresses which range from 1165 to 4000 lb. at different speeds and cut-offs.

By using heat-treated steel in the reciprocating parts and by the adoption of hollow piston-rods and by other savings in weight (but maintaining proper strength), the engine balances so that at 70 m.p.h. the variation allowable in pressure on the rail is less than $\pm$ 30 per cent. of the static load on the drivers.[2]

QUESTIONS ON CHAPTER XI

1. What device is applied to prevent burning out the superheater elements while no steam is passing through the superheater?
2. How does "gain in volume" compare in per cent. with "heat added to superheat" at 200 lb. pressure and 200°F. superheat?
3. Explain the action of superheated and saturated steam in the cylinder.

[1] Engineering, June 18, 1915.

[2] The principal locomotive ratios of the E6s are given under "4–4–2" in Table XXII, p. 225.

4. Would superheated steam give more work if it expanded according to the same law as does saturated steam? Show sketch.

5. Make a simple entropy diagram for superheater steam comparing the losses with saturated steam.

6. Does theory favor compounding saturated steam engines?

7. Which of the three methods named for superheating is of greatest practical advantage? Why?

8. Give a brief summary of the results of (*a*) road tests, (*b*) plant tests on basis of (1) coal, (2) water economy.

9. How do the cut-off and maximum power developed compare in superheater and non-superheater locomotives?

10. How does the best economy for water and for coal consumption in the superheater locomotive compare with stationary practice? (Consult reference books.)

(A problem in which tube surface is calculated, based on locomotive ratios for superheated steam, is given in the next chapter.)

CHAPTER XII

LOCOMOTIVE RATIOS

The Problem.—Based on careful design, after long experience and from trustworthy data, the engineer may produce a locomotive of exceptional merit; but if that locomotive has not been worked to advantage under the conditions for which it was built, or has been abused in shop or roundhouse, the extra attention in design has counted for little. It must not all be left to the designer to get results, but in any event certain ratios must be a guide or standard with which other results may be compared. In the following table, the values are favorable averages for present-day conditions.

TABLE XX

Evaporation per sq. ft. firebox heating surface	55 lb.
Evaporation per sq. ft. tube heating surface	7.5–14 lb.
Equivalent evaporation per sq. ft. heating surface (best)	16.4 lb.
Equivalent evaporation per sq. ft. heating surface (average)	10–12 lb.
Ratio length tubes to diameter	1–100
Ratio firebox heating surface to total heating surface	7–100
Horse power (including auxiliaries) for piston speeds 700 to 1000 ft. per min., per lb. saturated steam (best)	14.6 lb.
Same (average)	25–29 lb.
Average hp. per lb. saturated steam (simple)	27 lb.
Average hp. per lb. saturated steam (comp.)	23.5 lb.
Average hp. per lb. superheated steam (simple)	21.0 lb.
Average hp. per lb. superheated steam (comp.)	19.7 lb.
Dry coal (bituminous) per i.hp. hr. (best)	1.8 lb.
Dry coal (bituminous) per i.hp. hr. (average)	2–4 lb.
Dry coal (bituminous) per sq. ft. grate per hr.	100–120 lb.
Dry coal (hard) per sq. ft. grate per hr.	55– 70 lb.
Smoke box temperature (short tube)	750–800°F.
Smoke box temperature (long tubes)	550–600°F.

Designing Heating Surface.—In Chap. I, attention was directed to the necessity of properly balancing the boiler, cylinder, and tractive power. The common method of designing locomotive heating surfaces based on cylinder volume, is not consistent in the light of results from careful tests of locomotive

performance. Based on a study of recent experimental data and results, Mr. F. J. Cole, Chief Consulting Engineer American Locomotive Co., recommends in their pamphlet No. 1017 the use of cylinder horse power as the basis for heating surface and for grate and tube areas. His method of using locomotive ratios will be given with problems completely worked, showing their application to saturated and to superheated steam locomotives.

TABLE XXI.—EVAPORATION FROM TUBES AND FLUES IN POUNDS OF STEAM PER HR. PER SQ. FT. OUTSIDE SURFACE

Length in ft.	2″ Tubes				2¼″ Tubes				5½″ and 5⅜″ Flues			
	Spacing				Spacing				Spacing			
	¾″	1 3/16″	⅞″	1 5/16″	¾″	1 3/16″	⅞″	1 5/16″	1 1/16″	¾″	1 3/16″	⅞″
14	10.36	10.60	10.82	11.06	10.88	11.10	11.30	11.50	11.97	12.04	12.10	12.17
15	9.97	10.21	10.42	10.65	10.51	10.70	10.92	11.10	11.59	11.65	11.72	11.78
16	9.60	9.83	10.05	10.26	10.15	10.33	10.55	10.72	11.23	11.28	11.35	11.40
16½	9.42	9.65	9.87	10.07	9.97	10.15	10.37	10.53	11.05	11.11	11.17	11.23
17	9.27	9.48	9.69	9.89	9.80	9.98	10.19	10.34	10.88	10.94	11.00	11.06
17½	9.11	9.31	9.52	9.71	9.63	9.82	10.02	10.17	10.71	10.77	10.83	10.89
18	8.95	9.15	9.35	9.54	9.46	9.66	9.85	10.00	10.55	10.60	10.66	10.72
18½	8.79	8.99	9.13	9.37	9.29	9.48	9.68	9.83	10.39	10.44	10.50	10.56
19	8.63	8.83	9.02	9.20	9.12	9.30	9.51	9.66	10.23	10.29	10.35	10.40
19½	8.47	8.67	8.83	9.04	8.97	9.15	9.34	9.50	10.08	10.14	10.20	10.25
20	8.32	8.51	8.70	8.88	8.83	9.00	9.18	9.34	9.94	10.00	10.05	10.10
21	8.02	8.20	8.40	8.56	8.55	8.73	8.89	9.03	9.66	9.72	9.77	9.83
22	.....	.....	.....	.....	8.28	8.45	8.61	8.76	9.39	9.44	9.50	9.55
23	.....	.....	.....	.....	8.03	8.20	8.35	8.49	9.13	9.18	9.24	9.29

METHOD OF USING LOCOMOTIVE RATIOS (A. L. CO.)

(*a*) From weight limitation on drivers, and from service, type, etc., obtain the required tractive effort.

(*b*) From tractive effort, boiler pressure, stroke, and size of driving wheels, obtain diameter of cylinder. Ascertain horse power from diameter of cylinder and boiler pressure. Eq. 15 (Chap. III).

(*c*) Estimate total steam per hour from
hp. × 27.0 lb.—saturated steam.
hp. × 20.8 lb.—superheated steam.

(*d*) Estimate total coal per hour from
hp. × 4.00 lb.—saturated steam.
hp. × 3.25 lb.—superheated steam.

(*e*) Ascertain size of grate from total coal divided by 120, or hp. divided by
30.00 for saturated steam.
36.90 for superheated steam.

(*f*) The evaporation of the fire-box equals fire-box heating surface × 55 lb.

per sq. ft. per hr. If combustion chamber or arch tubes are used, add their heating surface to the fire-box.

(*g*) Subtract (*f*) from (*c*) to obtain tube and flue evaporation required.

(*h*) Obtain evaporative value of each tube or flue for length, diameter and spacing. Subtract total flue evaporation from (*g*) if boiler has superheater, and divide remainder by value for each tube to obtain number required. See Table No. XXI.

(*i*) To obtain percentage of boiler divide total pounds of steam proposed boiler will evaporate by pounds of steam required.

(*j*) When the proportions of existing boilers are desired for comparison with their engine cylinder horse power, or with other boilers, the evaporation value of the tubes can be obtained by multiplying their outside heating surface in square feet by the value in Table No. XXI for length, diameter and spacing.

Problem 26. *Saturated Steam.*—A 4-6-2 type with 150,000 lb. on driving wheels, 200 lb. boiler pressure, 75-in. drivers, 28-in. stroke, simple cylinders. Find number of tubes.

Assuming 33,600 lb. tractive power, factor of adhesion equals 150,000 divided by 33,600 or 4.46.

From Eq. 10, diameter of cylinder $= \sqrt[2]{\frac{33600 \times 75}{0.85 \times 200 \times 28}} = 23$ or direct from Table III, Chap. III.

(*b*) Horse power from Eq. 15 = 1764.

(*c*) Total steam per hr. = 1764×27 = 47,630 lb.

(*d*) Total coal per hr. = $1764 \times 4 = 7056$ lb.

(*e*) Grate area in sq. ft. = $1764 \div 30 = 58.8$ sq. ft.

(*f*) Fire-box, assumed 212 sq. ft., evaporation at 55 lb. = 11,660 lb.

(*g*) Leaving to be evaporated by tubes 35,970 lb.

(*h*) Surface of one 2-in. tube, 20 ft. long, after deducting for tube-sheets = 10.423 sq. ft.

Tubes 2 in. diam., 20 ft. long, spaced ¾ in.—rate of evaporation = 8.32. (Table XXI.)

Evaporation for each tube = $10.423 \times 8.32 = 86.7$.

Number of tubes required = $35{,}970 \div 86.7 = 415$.

Problem 26a. *Superheated Steam.*—A 4-6-2 type with 150,000 lb. on drivers, 200 lb. boiler pressure, 75-in. drivers, 28-in. stroke, simple cylinders. Find number of tubes.

Assuming 33,600 lb. tractive power, factor of adhesion equals 150,000 divided by 33,600 or 4.46.

From Table III, diameter of cylinder = 23.

(*b*) Horse power from Eq. 16. = 1904.

(*c*) Total steam per hr. = 1904×20.8 = 39,600 lb.

(*d*) Total coal per hr. = $1904 \times 3.25 = 6188$ lb.

(*e*) Grate area in sq. ft. = $1904 \div 36.90 = 51.6$ sq. ft.

(*f*) Fire-box, assumed 212 sq. ft., evaporation at 55 lb. = 11,660 lb.

(*g*) Leaving to be evaporated by tubes and flues 27,940 lb.

Tubes 2 in. diam., 20 ft. long, spaced ¾ in.—rate of evaporation = 8.32. (Table XXI.)

(*h*) Flues 5⅜ in. diam., 20 ft. long, spaced ¾ in.—rate of evaporation = 10.00.

Surface of one 2-in. tube, after deducting for tube sheets = 10.423 sq. ft.

Surface of one 5⅜-in. flue, after deducting for tube sheets = 28.011 sq. ft.

Assuming 30, 5⅜-in. flues, the evaporation is $30 \times 28.011 \times 10.00 = 8430$ lb.

27940—8403 = 19,537 lb. for tubes to evaporate.

Number of tubes = $19{,}537 \div (10.423 \times 8.32) = 225$.

Problem 27.—Referring to the dimensions of locomotive, Table XVIII, Chap. XI: Calculate as in Problem 26a, the leading dimensions for a superheater engine if *made over from the non-superheater* engine No. 1136, the cylinder dimensions and drawbar pull to be as in No. 1136. Compare your results with superheater engine No. 1160 in same table.

Fig. 98 presents a group of curves which may be used for approximate results in the solution of many problems for both saturated and superheated steam. The quadrants are arbitrarily designated as **1, 2, 3, 4.** Curves in Quadrant **1,** are plotted from Eq. 13, Chap. III, i.hp. = $T \times V/375$, where T becomes the available traction force for velocities V m.p.h. In quadrant **2,** the total evaporation per hour is the product of the i.hp. curves and the corresponding point on the inclined line marked "pounds of steam/i.hp. hr." For example, the total evaporation, if 1000 hp. is being developed with 20 lb. steam per i.hp. hour, would give 20,000. The same general method of construction applies to quadrants **3** and **4.** The following example will make clear the application of this diagram.

Problem 28.—A certain 4-4-2 type locomotive has a tractive effort of 10,900 lb. at 40 m.p.h. It requires 24 lb. of water per i.hp. hour from a feed temperature of 60°F. and it evaporates 7.5 lb. of water per lb. of coal, under a steam pressure of 200 lb. Find the grate area, if the ratio of heating to grate area is 70. From Eq. 13, or from quadrant **1,** i.hp. = 1160. From Table XVI, Chap. X, the factor of evaporation is 1.207. $7.5 \times 1.207 = 9.0$ (approx.) equivalent evaporation. In quadrant 1, locate point B corresponding with above and follow the dotted lines B–C–D–E–F–G–H–I for the several values. From C, reading to right to line 24 and down to E, we find that 27,840 lb. water will be evaporated. Continuing down to line 9 and to the left to G, gives the total heating surface (based on fireside of tubes) of 3100 sq. ft., continuing to line 70 and thence up to "grate surface," we find 43 sq. ft. to be the answer.

Problem 29.—Find the total water evaporated and the pounds of steam per i.hp. hour of a locomotive with grate surface 55 sq. ft., ratio of heating to grate surface 50, equivalent evaporation 10 lb. per sq. ft. heating surface, drawbar pull 15,000 lb. at 30 m.p.h.

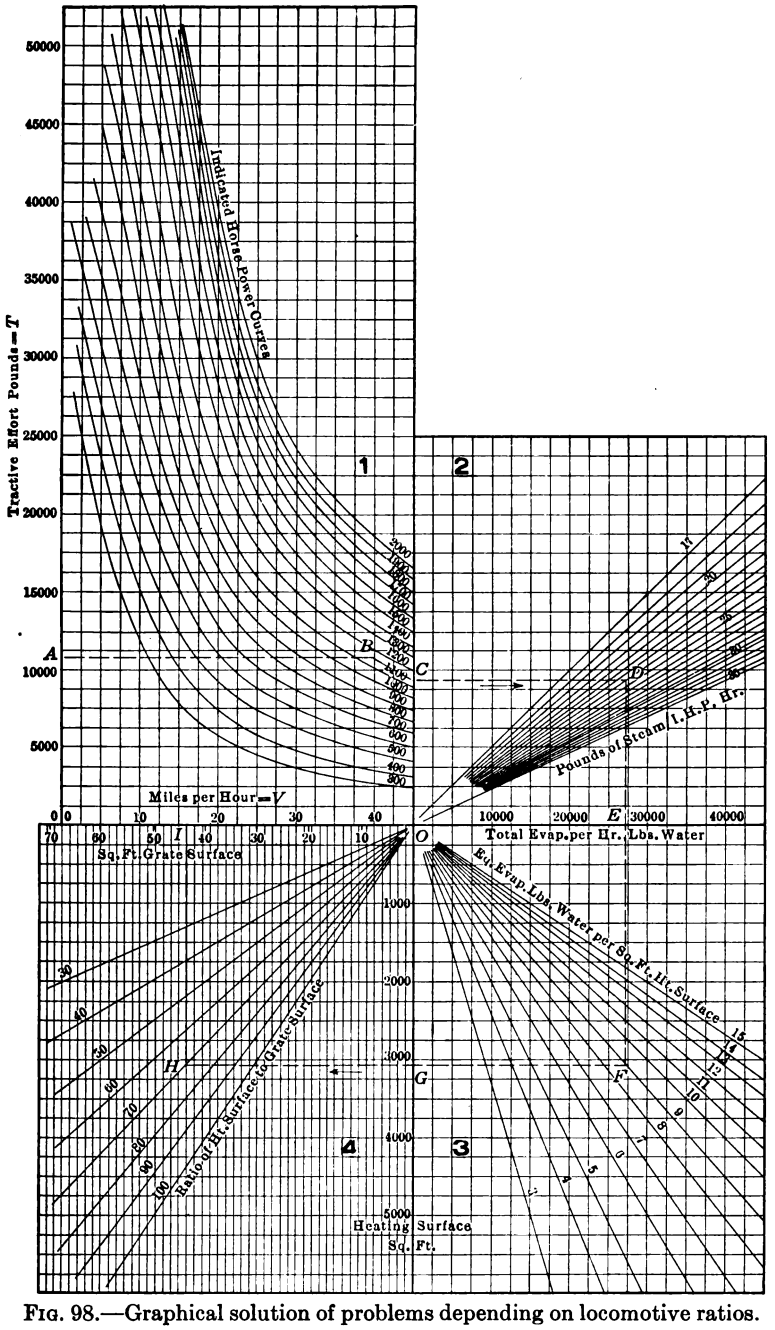


FIG. 98.—Graphical solution of problems depending on locomotive ratios.

Significant Ratios.—The following ratios, based on values deduced by Mr. Lawford Fry, were presented by Mr. L. R. Pomeroy in the American Engineer, Feb., 1913.

Let:

m.p.h. = maximum speed at which a locomotive can deliver its greatest tractive force.

H.S. = total heating surface in sq. ft.

T.E. = rated tractive force or effort.

x = constant depending on rate of evaporation.

w = weight cu. ft. steam at assumed m.e.p.

P = boiler pressure.

$$\text{Total steam per hour} = \frac{36.65 \times \text{m.p.h.} \times \text{T.E.} \times w}{0.85P}$$

$$\text{m.p.h.} = \frac{x \times \text{H.S.}}{\text{T.E.}} \qquad (97)$$

	Saturated steam	200° F. superheated steam
For an evaporation of 8 lb. per sq. ft. H.S., x =	99.4	130
For an evaporation of 9 lb. per sq. ft. H.S., x =	112.0	149
For an evaporation of 10 lb. per sq. ft. H.S., x =	124.0	166
For an evaporation of 11 lb. per sq. ft. H.S., x =	137.0	184
For an evaporation of 12 lb. per sq. ft. H.S., x =	149.0	198

Maximum coal per hour = T.E. × m.p.h. × Z

	Saturated steam	Superheated steam
For 7 lb. water per lb. coal, Z =	0.01150	0.00866
For 8 lb. water per lb. coal, Z =	0.01006	0.00758
For 9 lb. water per lb. coal, Z =	0.00894	0.00673

Problem 30.—Find the value of m.p.h. in Eq. 97 for each of the locomotives in Table II, page 12.

Fig. 99 gives results of tests on a 4–4–0 type, class E2*a* locomotive, showing the relation of drawbar pull, drawbar horse power, water evaporated and dry coal fired. For example, if this locomotive is required to haul a train at 160 r.p.m. with a drawbar pull of 7000 lb. behind engine, it would consume 3600 lb. of coal and 25,000 lb. of water per hr. and the drawbar horse power would be 750. This diagram gives a summary of test results, showing the influence of speed on an Atlantic type (dimensions as in Table II, Chap. II) and the results are fair averages for a non-superheater engine of the class tested.

For reference in design, the following diagrams of locomotive ratios are arranged from Henderson's "Locomotive Operation."

Fig. 100 shows the maximum evaporation, pounds of water per sq. ft. of heating surface per hr. from and at 212°, which can be obtained under the most favorable conditions.

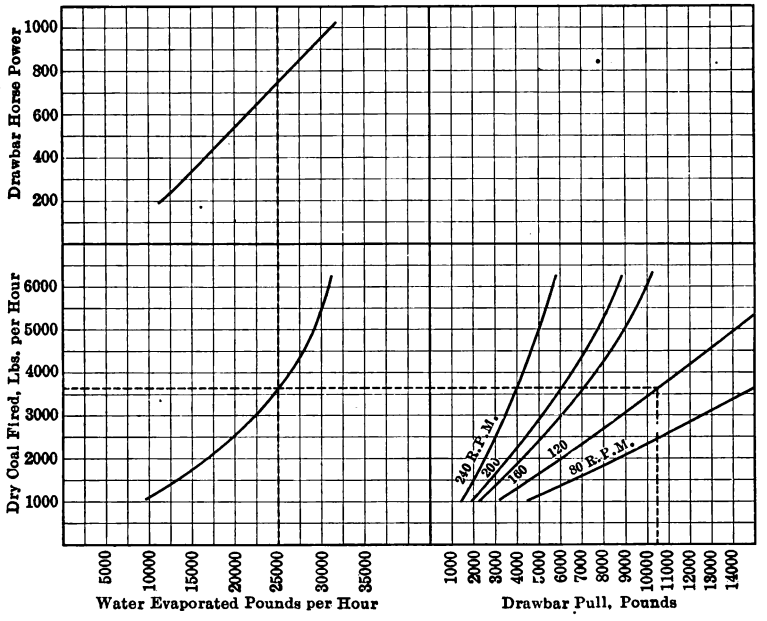


FIG. 99.—Ratios from test on a 4–4–2 (E2a) locomotive.

Curve "*a*," large sizes of anthracite coal.
Curve "*b*," small sizes of anthracite coal.
Curve "*c*," Pennsylvania and Virginia semi-bituminous coal.
Curve "*d*," Indiana and Illinois bituminous coal.
Curve "*e*," fuel oil.

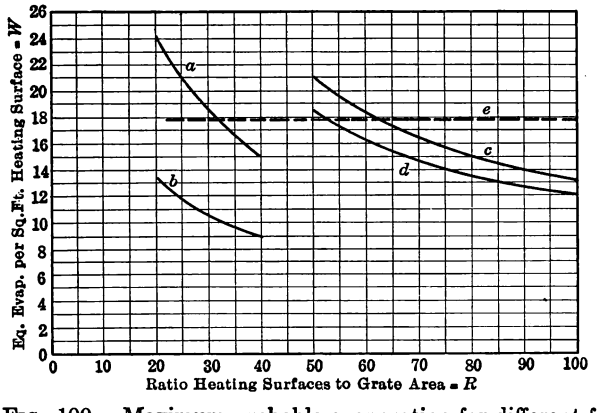


FIG. 100.—Maximum probable evaporation for different fuels.

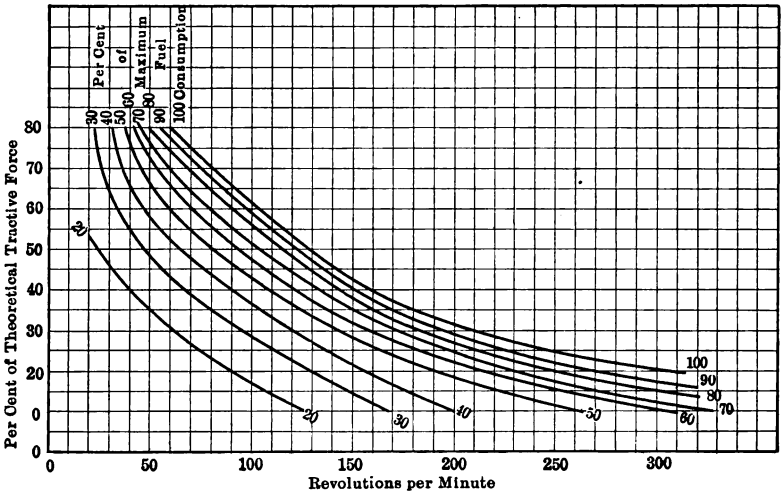
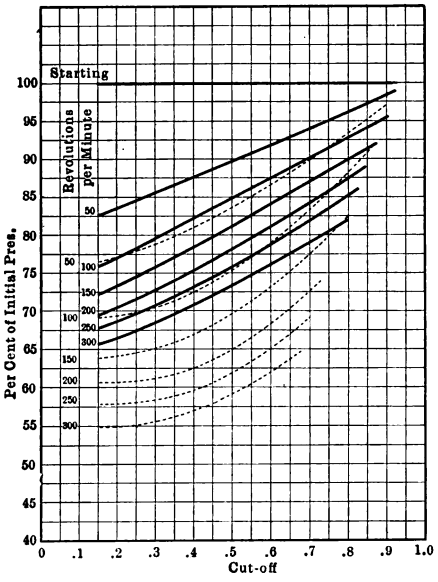


FIG. 101.—Ratio of fuel consumption to maximum combustion. Available tractive force taken as 80 per cent. theoretical.



Heavy lines are for upper limit, as for double-ported valves.
Dotted lines are for lower limit.

FIG. 102.—Ratio of cut-off to initial (gage) pressure.

Fig. 101 brings out the effect of different rates of combustion in per cent. of the maximum, on the theoretical tractive force. These results are from the Chicago & Northwestern taken some years ago, but which check well with some recent tests.

Fig. 102 shows the effect of increased speed on the cut-off pressure. Suppose a locomotive with 68-in. driving wheels is running at 30 m.p.h., with boiler pressure 200 lb. and cut-off 65 per cent. From Table XXVII, appendix, the r.p.m. at this speed is 148 and for this speed Fig. 102 gives (light lines) 75 per cent. cut-off to initial pressure. This percentage will be higher if the port-openings are made larger.

Table XXII gives a list of ratios commonly referred to in comparing different locomotives.[1] The columns to the right contain values for a few modern superheater locomotives.

TABLE XXII.—RATIOS

	4-4-0	4-4-2	4-6-0	2-8-8-8-2
1. Road	P. & R.	P.R.R.	D. & H.	Erie
2. Builder	Baldwin	P.R. Co.	Amer.	Baldwin
3. Coal	Anth.	Bit.	Bit.	Bit.
4. $\frac{\text{Weight on drivers}}{\text{Tractive effort}}$	4.32	4.52	4.68	4.76
5. $\frac{\text{Total weight}}{\text{Tractive effort}}$	6.23	8.15	6.31	5.33
6. $\frac{\text{Equiv. heating surface}^2}{\text{Grate area}}$	22.1	71.3	59.3	102.9
7. $\frac{\text{Fire box heating surface}}{\text{Equiv. heating surface}^2}$	11.56 %	4.93 %	5.09 %	3.89 %
8. $\frac{\text{Weight on drivers}}{\text{Equiv. heating surface}^2}$	63.4	33.8	49.7	82.2
9. $\frac{\text{Total weight}}{\text{Equiv. heating surface}}$	91.3	59.7	67.2	92.1
10. $\frac{\text{Equiv. heating surface}^2}{\text{Vol. of cylinders (cu. ft.)}}$	197.0	300.0	238.0	180.5
11. $\frac{\text{Grate area}}{\text{Vol. cylinders}}$	8.94	4.21	4.02	1.75

[1] For tabulation of 36 recent locomotives, giving complete data, see Railway Age Gazette, Mechanical Edition, June and August, 1915.

[2] Equivalent heating surface = total evaporative heating surface + 1.5 times superheating surface.

CHAPTER XIII

LOCOMOTIVE TESTING

Outline for Road Tests and Location of Instruments; Mounting for Laboratory Tests; Description of Brakes Which Absorb the Power

The purpose for which a test is conducted determines largely the instruments to be used and the observations to be made. No test should be undertaken without the most careful predetermination of conditions and the object of the test fixed clearly in mind. The methods to be employed, the accuracy of the instruments used, and the mental aptitude and integrity of the observers and calculators are of primary importance.

Road Tests.—The location of test instruments on a 4–8–0 locomotive equipped for road tests is shown in Fig. 103. The

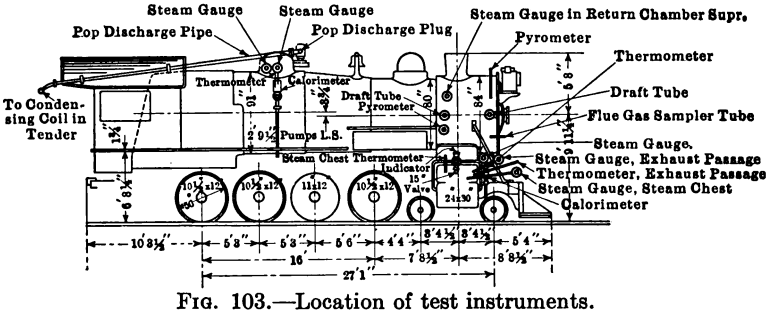


FIG. 103.—Location of test instruments.

following has been arranged from the report of "Committee on Steam Locomotives—Methods of Conducting Laboratory and Road Tests." Proceedings A. R. M. M. Asso., 1914.[1]

OBJECT.—The object of a *road test* is to determine the steam and coal consumption of a locomotive per unit of power under practical conditions of the locomotive in railroad service.

PREPARATIONS.—All of the preparations as given in laboratory tests[2] should be carried out preparatory to placing the locomotive in service, with

[1] Consult also Code 1914, Report of Power Test Committee, The American Society of Mechanical Engineers.

[2] This includes shop inspection and (where necessary) prevention of leaks, cleaning of boiler, "lining up," etc., referred to in the original report.

the possible exception of not having all driving wheels newly turned, and equipping the locomotive with the various instruments that can be done while the locomotive is in the shops for repairs.

FUEL.—The same consideration should be given to the fuel as on a laboratory test.

To facilitate the measurement of coal and the determination of the quantity used during any desired period of the run, it is desirable to provide sufficient number of sacks, of a size holding 100 lb., and to weigh the coal into these sacks preparatory to starting on the test.

APPARATUS AND INSTRUMENTS.—The apparatus and instruments required for a road test of a locomotive are as follows:

No. 1. Platform scale for weighing coal.

No. 2. Crane, spring balance and bucket for weighing ash.

No. 3. Tank and scales for calibrating the tank.

No. 4. Graduated scale attached to water glass on boiler.

No. 5. Float for measuring height of water in tank, or, if preferred, graduated scales on all four corners of the tank.

No. 6. Pressure gages graduated to pounds for boiler, branch pipe, receiver and exhaust.

No. 7. Draft gages for smoke box, fire box and ash pan.

No. 8. Thermometers for calorimeter, branch pipe, receiver and exhaust.

No. 9. Pyrometers for fire box, smoke box, and at other points as required.

No. 10. Air-pump counters.

No. 11. Water meters.

No. 12. Steam calorimeter.

No. 13. Steam-cylinder indicators.

No. 14. Some form of speed recorder for the revolutions for the driving wheels in case no dynamometer is accessible. On Mallet type of locomotives two recorders should be used.

No. 15. Some form of pendulum-indicator rigging.

No. 16. Traction dynamometer for determining pull at drawbar, with its complete equipment.

No. 17. Electrical connection between locomotive and dynamometer.

In addition to the above it will be necessary to have planimeters, micrometers, scales and calculating machines, etc.

Steam used for auxiliary purpose other than the cylinders, such as air pump, calorimeter, injector overflow, train lighting and heating and what escapes from the safety valves, may be estimated from data obtained by testing them either before or after the trial.

The scales, gages and pyrometers should be calibrated before and after the tests are made.

APPLICATION OF INSTRUMENTS.—The indicator rig should be some form of pendulum motion with a light tube for transmitting the reduced motion to a point near the indicator.

The apparatus which is most suitable consists of a three-way cock for the attachment of the indicator, with a steam-chest connection, so that diagrams can be drawn on each cylinder card and pressure determined.

The three-way cock should be provided with a clamp rigidly secured

to the cylinder and thus overcome any tendency of the indicator to move longitudinally with reference to the driving rig. The support for the motion rod should be secured to some point on the steam chest. Care should be taken to set the indicators in such a position that the finger on the end of the motion rod travels in a direction pointing to a groove in the drum proper.

The pipes leading from the cock to the cylinder should be not less than ½ in. inside diameter, and if possible not exceeding 36 in. in length. They should be connected into the side of the cylinder, rather than into the heads. Sharp bends in the pipe should be avoided and they should be well lagged to reduce radiation.

If a dynamometer car is not used, stroke counter should be placed at some convenient point in the pilot box to record the revolutions of the drivers. This can be conveniently driven from a finger on the motion rod of the indicator rigging.

To facilitate the working of the men who operate the indicators and read the instruments at the front of the locomotive, and to protect them from wind or rain and jolting, a suitable pilot box extending back to the cylinder and properly secured to the bumper beam should be provided.

Whenever practicable, the bulb of the thermometers used in branch pipe, receiver or exhaust should come in direct contact with the steam and no wells used. When thermometers are placed in wells, they do not respond quickly with the different changes in the working of the locomotive.

The water meters should be attached to the suction pipes of the injectors, and located at points where they can be conveniently read while the locomotive is in motion. Each meter should be provided with a check valve to prevent hot water from flowing through them from the injectors, and strainers to intercept foreign material. With the water scoops it will be impossible to use a float, but when tests are made on roads not using water scoops, a suitable float should be made for determining the water consumption. The water level may be established by using a rubber hose with glass tube inserted in the end, which will indicate the height of water in the tank, this tube to be brought in contact with a properly calibrated scale, or, if more convenient, long glass tubes may be provided at each corner of the tank for the same purpose.

In all cases the term "branch pipe" refers to the steam-supply pipe to the cylinders and not the injector branch pipe.

Duration.—The duration of a test is the running time minus time the throttle is closed, and depends upon the length of the run between locomotive terminals. In fast passenger service the runs should be, if practicable, at least 100 miles long. In service requiring frequent stops and in freight service, the distance may be much shorter. The length of time upon which the hourly rate of consumption and evaporation are based is the total time that the throttle valve is open and not elapsed time between the starting and stopping time.

Starting and Stopping.—The fire having been thoroughly cleaned, banked to permit coking, fresh fuel should be supplied to a level thickness which will be required for the run. After the locomotive is attached to the train, observe the pressure, the water level or meter readings, and when the locomotive starts take this as the starting time. Thereafter cover the fire

with weighed coal and proceed with the regular work of the test. The ashes and refuse should be removed from the ash pan and smoke box before the locomotive is coupled to the train.

During the run the fire should be maintained in as equal and uniform condition as practicable, and when the end of the route is reached the fire should be as level and approximately the same thickness and condition as at the start. When the locomotive is stopped and the proper level of the fire obtained, the weighed coal should be discontinued. If during the run a stop of over 7 min. is made, and in order to keep the fire in proper condition fresh fuel must be supplied, this should be selected from the unweighed coal. There should preferably be no water supplied to the boiler, and if it is supplied, allowance should be made for same.

On reaching the terminal, the fire being in the same condition as at the start, the water level and water supply should be noted. The time the locomotive comes to rest should be the time of stop of test.

In the dynamometer car at least four observers are required, one to record the time of each start and stop, passing each station and recording mile posts, point of curvature and tangent and any other important information; one to record all information on the diagram and keep track of indicator cards, and one to take car numbers and weights of trains; this latter man can also act as a relief observer. When making test of Mallet type of locomotive, the locomotive force is increased to take indicator cards from the low-pressure cylinders.

The time to take records depends entirely upon what facilities are available for recording same. If a dynamometer car is available for the tests, records should only be taken when some change in the operation of the locomotive takes place, such as throttle lever, reverse lever and boiler pressure. If the dynamometer car is not available, all records should be taken preferably every 5 min.

Special reading of the meters and total number of sacks of coal fired should be taken at specified stopping and passing points.

Careful observations should be made throughout the run, of the time passing all important points, arriving and leaving each station, and the time that the throttle valve is opened or closed, not only at each stop, but when drifting.

Ash and Refuse.—In weighing and sampling the ash and refuse, the same preparation as described for laboratory tests should be followed as far as practicable.

Sampling Coal.—The coal should be sampled while it is being weighed off in 100-lb. lots, and a small proportion taken at different times until about 300 lb. is obtained. This should be crushed and quartered and about 1 qt. placed in an air-tight jar and sent to chemist for analysis. When this method of sampling is used, care should be taken that the coal does not take on additional moisture, due to leaky cistern or sprinkler. If there is any question as to the coal taking additional moisture after it is once weighed out, sample should be taken from each sack as they are emptied.

On all tests the total moisture should be used in all calculations.

Calorific Test of Coal.—The "proximate" analysis discussed in Chap. IX meets the requirements of road tests.

TABLE XXIII.—*Data and Results.* The data and results should be reported in accordance with the form given for laboratory tests as far as practicable, and in addition a summarized form should be made giving the following information.[1]

The item number given first (as "793") is the number appearing in the original report and corresponds with the number in the form for laboratory tests.

793. [1]Date of test.....
794. [2]Average number cars (pushed or pulled).....
795. [3]Gross tons, excluding locomotive.....
796. [4]Number 100 gross ton-miles.....
797. [5]Number 100 adj. ton-miles.....
798. [6]Number of stops.....
799. [7]Distance in miles.....
800. [8]Time of trip.....
801. [9]Time running—hours.....
802. [10]Time throttle open—hours.....
803. [11]Average speed, running throttle open—m.p.h.....
380. [12]Average boiler pressure, lb. per sq. in.....
384. [13]Average hp. steam-chest pressure, lb. per sq. in.....
385. [14]Average L. P. steam-chest pressure, lb. per sq. in.....
394. [15]Draft, front of diaphragm—in. water.....
395. [16]Draft, back of diaphragm—in. water.....
373. [17]Temperature feed-water—Degrees Fahr.....
368. [18]Temperature air—Degrees Fahr.....
409. [19]Degrees superheat in branch pipe—Degrees Fahr.....
410. [20]Degrees superheat in receiver—Degrees Fahr.....
411. [21]Degrees superheat in exhaust—Degrees Fahr.....
804. [22]Coal and how fired.....
418. [23]Coal, total as fired, lb.....
626. [24]Dry coal per hr. fired, lb.....
627. [25]Dry coal fired per hr. per sq. ft. grate area.....
805. [26]Water, total out of tender, lb.....
806. [27]Water, total evaporated, lb.....
638. [28]Water loss—Calorimeter, safety valves, etc., lb.....
637. [29]Water evaporated per lb. fuel as fired.....

EQUIVALENT EVAPORATION FROM AND AT 212°F.

645. [30]Per hr. lb.....
648. [31]Per hr. per sq. ft. total heating surface.....
657. [32]Per hr. per lb. coal as fired.....
660. [33]Boiler hp. (34.5 U of E).....
666. [34]Efficiency of boiler based on fuel—per cent.....
639. [35]Dry steam to engines—lb. per hr.....
722. 723. [36]I.hp., high-pressure cylinders.....
724. 725. [37]I.hp., low-pressure cylinders
728. [38]Total i.hp.....

734. [39]Dry coal per i.hp. hr.
736. [40]Dry steam per i.hp. hr.
487. [41]Average drawbar pull, lb.
743. [42]Dynamometer or drawbar horse power
744. [43]Dry coal per d.hp. hr.
745. [44]Dry steam per d.hp. hr.
807. [45]Coal as fired per 100 gross ton-miles, lb.
808. [46]Coal as fired per 100 ton-miles, lb.
809. [47]Water per 100 ton-miles, lb.
810. [48]Coal as fired per car-mile, lb.
811. [49]Water per car-mile, lb.

The results of a complete road test by Prof. E. A. Hitchcock, are given in Transactions A. S. M. E., Vol. XXVI.

Laboratory Tests.—Frequent references in former chapters have been made to results of tests from the Purdue University and from the Pennsylvania Railroad locomotive testing plants. More recently, a plant designed by Prof. E. C. Schmidt, adopted for testing the largest locomotives built, has been completed for the University of Illinois.

It will be understood that the chief advantage of the testing-plant method over that of road tests lies in the possibilities of keeping constant conditions during a test and this, in turn, makes it possible to accurately compare results with those obtained under other test conditions.

Fig. 104 shows the general plan of mounting a locomotive, and the frontispiece of this book illustrates a large Pacific type engine under tests at the Altoona plant. The energy from the steam driving the locomotive, which normally is absorbed by the mass of the moving train, is absorbed by brakes keyed to large supporting wheels. These wheels, arranged for testing a 4–4–2 engine, are shown as *J*, Fig. 104.

The axles of the driving wheels are extended and each is equipped with a set of Alden absorption brakes, a section of one being shown in Fig. 105. The discs D are a part of the hub which is keyed to the supporting axle. On each side of the revolving discs are copper diaphragms CC which are forced against the discs by the pressure of water in the spaces W. These surfaces in contact are lubricated by oil entering near the hub and forced to the circumference by centrifugal action, where it escapes. The bearing surface between the hub and the housing is lubricated by the oil which leaks past the packing rings (e).

Water under pressure circulating through the spaces W press

the copper plates against the revolving discs and introduce the resistance to turning. The copper plates and the discs are protected from wear by the oil, the resistance being due to the viscosity of the lubricant under the imposed conditions of pressure. The heat generated is carried away by the water.

The water enters the brake at the bottom and passes out at the top, the pressure and amount of water being controlled by an inlet valve for each pair of brakes. The brake housing is kept from turning by two tie rods,

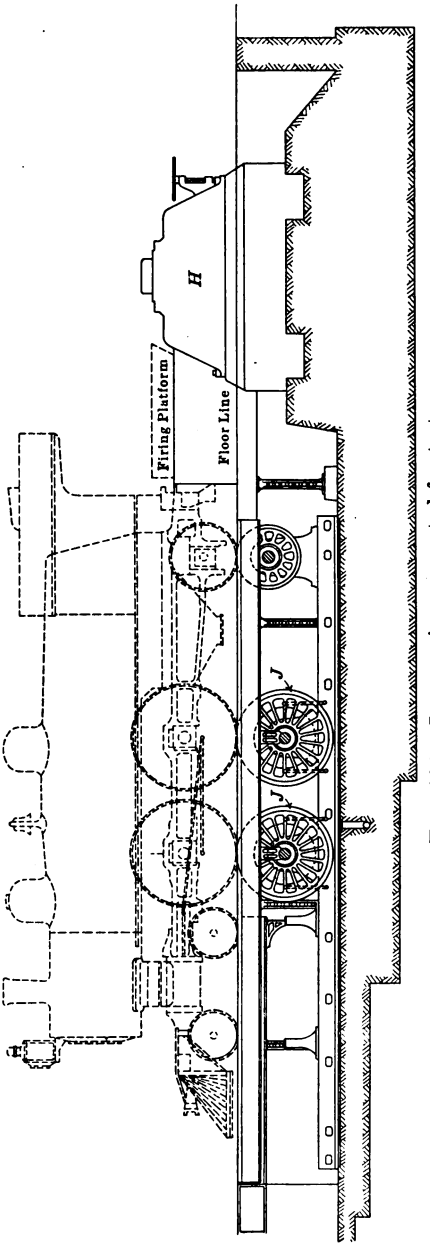


FIG. 104.—Locomotive supported for tests.

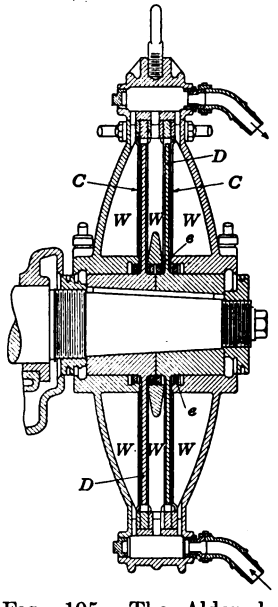


FIG. 105.—The Alden hydraulic brake.

the lower ends of which are bolted to brackets on the bed-plate casting. These tie-rod brackets are fastened to the bed-plates by bolts in T slots, so that the brakes can be moved to any desired position.

The water required for locomotives under test is weighed in two tanks, filled alternately, and run from these into a third tank, from which it is taken by piping to the connections for the injectors.

The traction dynamometer at H, Fig. 104, which measures the drawbar pull of the locomotive, is of the lever type and is constructed on the "Emery" principle, in which flexible steel plates take the place of knife edges used in ordinary scales, the same as in the dynamometer car, illustrated in Chap. VII. The weight of each lever is taken by a vertical plate in a plane intersecting that of the receiving fulcrum plates at their center of rotation, thus relieving these plates of all transverse load. The yoke embracing the dynamometer and to which the drawbar is attached, is also mounted on flexible plates and braced by long, flexible rods, to insure frictionless motion in the horizontal plane only.

The total motion of this yoke and drawbar, due to the leverage of the machine and to stress of parts when under full load, does not exceed 0.04 in., so that a locomotive exerting a drawbar pull equal to the full capacity of the dynamometer, will not move forward on the supporting wheels more than the amount specified. The drawbar is provided with a ball joint, to allow for any side motion of the locomotive, or motion of the locomotive on its springs.

Near the base of the dynamometer, the oscillating motion of the ends of the last levers is transformed into a rotary motion by means of steel belts wrapped around a drum and kept in constant tension by suitable clamping devices. The belt drum is mounted on a tube guided in ball bearings, and inside of it is a rod, the upper part of which is securely fastened to the tube, the lower end being firmly attached to the frame of the machine. It will thus be seen that when the belt drum is rotated, the rod inside of the tube is in torsion, and this resistance forms part of the total resistance of the machine, and is a constant for the same travel of the recording pen.

To the upper end of the tube mentioned, are secured two radial arms, the extreme ends of which are finished to a circle

having its center at the center of the tube. The angular motion at the end of one arm imparts straight line motion to a carriage, guided by a grooved track and carrying the recording pen. The opposite arm is coupled by steel belts to a rotary oil dash pot, to reduce violent oscillations of the recording pen, the extent of which can be controlled as desired. The principal resistances in the dynamometer are flat springs, placed under the second levers and deflected by the motion of these levers. There are three sets of these springs, varying in resistance so that a travel of 8 in. of the recording pen corresponds to a drawbar pull of either 80,000 lb., 40,000 lb. or 16,000 lb. as may be desired. The hydraulic type has advantages over the spring type for locomotive dynamometers.

The drawbar pull is traced upon a strip of paper 18 in. wide,

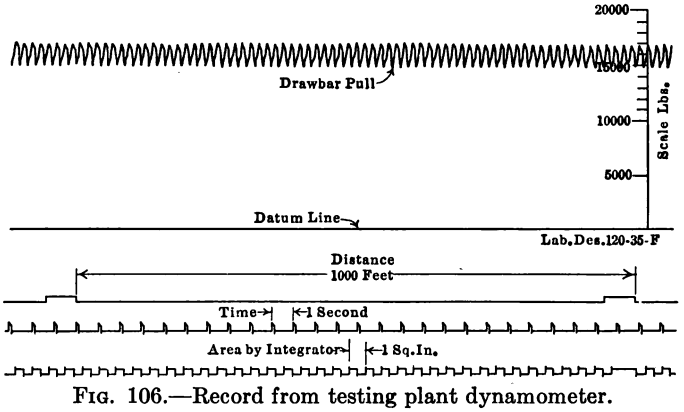


FIG. 106.—Record from testing plant dynamometer.

made to travel at a known rate for each mile run by the locomotive. The record shown in Fig. 106 was taken at 20 m.p.h. (120 r.p.m.) and under this condition, without dashpots in the safety bars, the *variation* in the horizontal forces in the drawbar pull was about 2000 lb., a mean line through this record, being the average pull. The *distance*, *time* and *integrator* records are used only when specially required, these being used as checks on other calculations. The ft.-lb. of *work done* is found by multiplying together the average drawbar pull, the average circumference of the driving wheels in feet and the total revolutions. The *dynamometer horse power* equals the product of the constant for power developed when the speed is 1 r.p.m. and the pull is 1 lb., the r.p.m., and average drawbar pull.

Other testing instruments include steam-engine indicators, steam pressure and draft gages, thermometers, pyrometers, calorimeters and revolution counter, the usual locations of which are shown in Figs. 103 and 107. With but few exceptions, the instruments and their application on the locomotive, are the same as given for road testing.

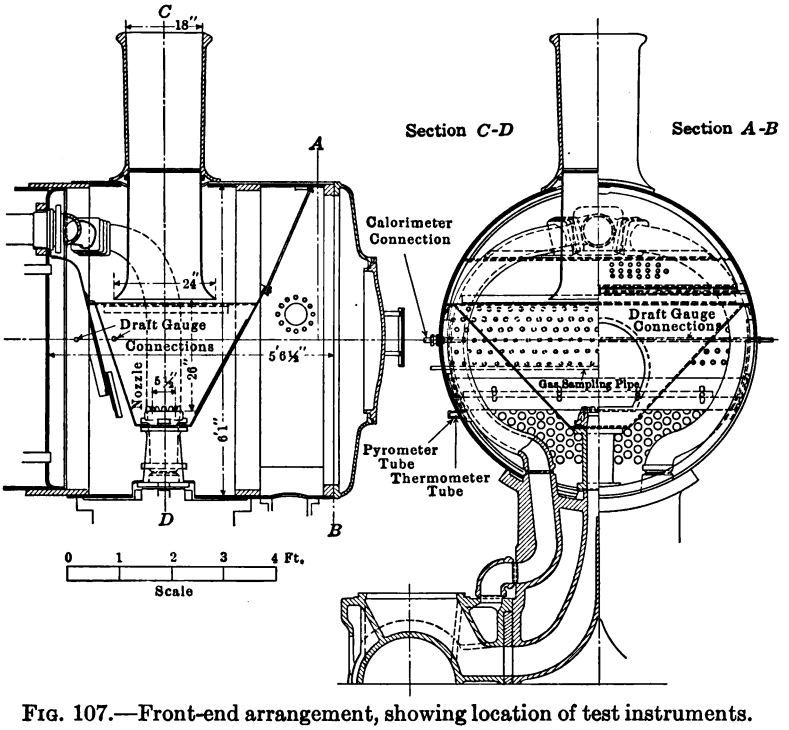


FIG. 107.—Front-end arrangement, showing location of test instruments.

Since 1912, speeds up to 85 m.p.h. have been reached in the regular tests on the P. R. R. plant. Based on new and important data obtained, locomotive design and operation has come to have a new significance. The student is urged to consult carefully the various bulletins giving results on locomotive testing plants; the references to these reports as made in this text are merely suggestive of the wide range of investigations undertaken.

CHAPTER XIV

COUNTERBALANCING

THEORY, BRIEFLY STATED; WHY IMPORTANT; COMMON METHODS; PROPOSED METHODS; PROBLEMS

If a weight of W lb. is moving at a uniform velocity of v ft. per sec. at a distance r ft. from a fixed point, the *force, F_c, in pounds,* which would constrain it to move in a circular path will be

$$F_c = \frac{Wv^2}{32.16r} \tag{98}$$

If ω is the angular velocity in radians per sec., $v = \omega r$ and if N = r.p.m., Eq. 98 becomes:

$$F_c = \frac{W\omega^2 r}{32.16} = 0.00034WN^2r \tag{98a}$$

If W is the revolving weight (as the crank pin) on a driving wheel, it can be perfectly balanced simply by applying a balancing weight on the same diameter. If the force due to the inertia of the reciprocating parts is transmitted to the crank pin, we must apply some "overbalance" weight to neutralize the inertia effects of these parts. At the best, this can be only an approximation, for the forces vary with the square of the velocity (as shown in above equations) and with the position of the cross-head. If the horizontal component of the unbalanced forces are not neutralized, a "fore and aft" motion is given successively to the frame on either side, producing an effect called "nosing." Cushioning the steam near the end of the stroke, when the inertia force is at its maximum, may somewhat reduce this effect but cannot eliminate it. When the unbalanced weight, or any component of it, acts vertically, the rail pressure is increased if below the center line and decreased if above. At high speeds, imperfect balancing may cause the driving wheels to leave the rail and then return with a "hammer blow" sufficient to force it out of alignment or to break a rail. During high-speed tests on a locomotive testing plant, it is necessary to balance close to 100 per cent. If it were possible to design curves on the

right of way which would constitute the run of a particular locomotive, the counterbalancing of such locomotive would be largely based upon the speed of its travel, which would be worked out by the transportation department and the curves and superelevation would have to be worked up in conjunction therewith. As there is a large variation of linear velocity for the different classes of locomotives operating over the same right of way, a compromise must be worked up for the counterbalance weights for each class.

Principles in Locomotive Counterbalancing.[1]—The complete solution of this problem, based on both theory and practice, requires extended analysis, but the general principles can be grasped by a study of Figs. 108 to 111.[2]

The radius of the circle, in each case, represents the centrifugal force of the overbalance, which is the centrifugal force of the weight added to partly counterbalance the reciprocating parts.

The diagrams represent conditions at high speed.

The revolving parts may be assumed to be perfectly balanced, so that the weight added for that purpose is not represented in these diagrams. The weight added for partly balancing the reciprocating parts is the *overbalance* which distorts the otherwise perfectly balanced revolving parts. This overbalance is represented in the four diagrams by parts shown black.

Fig. 108 represents, by shaded and unshaded portions, the total weight of reciprocating parts, the shaded portion within the circle being balanced by the overbalance when the wheel is in the position shown. The portion without the circle is the unbalanced weight of reciprocating parts, which tends to cause nosing, that is, a fore and aft irregular movement of the locomotive.

Fig. 109 shows the position of the wheel after a quarter-turn, in which the effect of the unbalanced reciprocating parts is eliminated, and the distorting forces are caused by the centrifugal force of the overbalance acting in a downward direction, the resultant effect on the track being the static weight on the driving wheel plus the centrifugal force of the overbalance. This position gives the greatest pressure on the rail.

[1] Arranged from 1915 report of Committee on "Counterbalancing," A. R. M. M. Asso.

[2] One of the best discussions of the principles of counterbalancing is by Professor W. E. Dalby in his treatise "The Balancing of Engines." See also Vols. XXIX and XXX, A. R. M. M. Asso., containing report (with discussion) of Committee on Counterbalancing.

Fig. 110 is similar to Fig. 108, and shows the effect of unbalanced reciprocating parts in the opposite direction, after another quarter-turn. There are, however, slight differences in the effect shown in Figs. 108 and 110, due to angularity of connecting rods, etc., but this need not be considered.

Fig. 111 shows the downward force on the track when crank pin is down and overbalance is up, this force being the difference

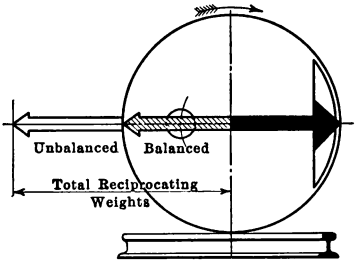


FIG. 108.

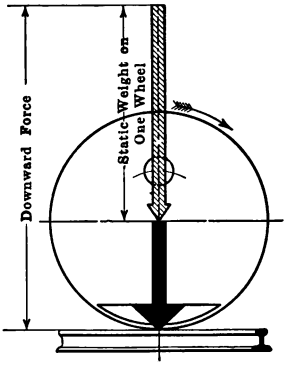


FIG. 109.

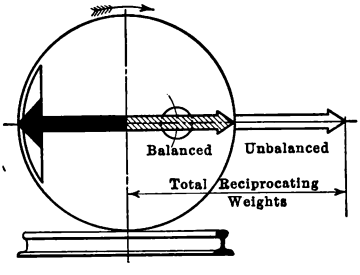


FIG. 110.

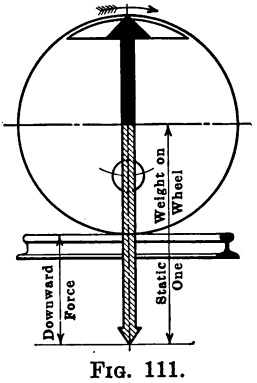


FIG. 111.

Illustrating principles of counterbalancing.

between the static weight on driver and centrifugal force of the overbalance. This position gives the least pressure on the rail. The proportions in this figure show the overbalance to neutralize about one-half the static weight on the wheel, leaving half the static weight as the downward force on the track for this position of the crank.

The overbalance which is used to counteract the desired por-

tion of the weight of the reciprocating parts should be distributed as nearly equally as possible among all driving wheels, adding to it the weight of the revolving parts for each wheel. This sum for each wheel, if placed at a distance from the driving wheel center equal to the length of the crank, or a proportionally less weight if at a greater distance, will be the counterbalance required.

Cross-counterbalancing, to correct the disturbances caused by the parts revolving in different planes, is thought to be unnecessary with outside cylinders, on account of the disturbing forces being slight when compared to the principal reciprocating and centrifugal forces.

Centrifugal and reciprocating forces are usually figured at a speed in miles per hour equal to the diameter of the driving wheel in inches, which may be considered as a maximum for good practice. This is ordinarily referred to as "diameter-speed." At this speed the reciprocating parts, due to the laws of inertia, tend to continue their motion at the end of each stroke with a force (acting on the crank pin) about equal to 40 times their weight. This force, which may be termed the "dynamic augment," may be expressed in equivalent units by centrifugal force, in which case N (in Eq. 98*a*) becomes $\frac{5280 \times 12 \times V}{\pi D \times 60} = 336.13 \frac{V}{D}$, when V = velocity, m.p.h. and D = diameter of driving wheel in inches. At diameter speed, this gives approximately $N = 336$, which substituted in Eq. 98*a*,

$$F_c = 38.4Wr \qquad (98b)$$

If $r = 1$, the force is about 40 times the weight. Eq. 98*b* reduces to,

$$F_c = 1.6Ws = 3.2\ Wr \qquad (98c)$$

where s is the stroke in inches, r the crank radius, in inches, and W is the over-balance weight.[1] The overbalance exerts a

[1] If W = overbalance or excess weight at stroke distance,

then with 18-in. stroke, dynamic augment = 28.8 × W at diameter-speed.
then with 20-in. stroke, dynamic augment = 32.0 × W at diameter-speed.
then with 22-in. stroke, dynamic augment = 35.2 × W at diameter-speed.
then with 24-in. stroke, dynamic augment = 38.4 × W at diameter-speed.
then with 26-in. stroke, dynamic augment = 41.6 × W at diameter-speed.
then with 28-in. stroke, dynamic augment = 44.8 × W at diameter-speed.
then with 30-in. stroke, dynamic augment = 48.0 × W at diameter-speed.
then with 32-in. stroke, dynamic augment = 51.2 × W at diameter-speed.
then with 34-in. stroke, dynamic augment = 54.4 × W at diameter-speed.

centrifugal force equal to about 40 times its weight, and is at a maximum at the top and bottom position of the crank. This force is added to the static weight, in the lower position of the overbalance, and is opposed to this weight, in the upper position, as shown in Figs. 109 and 111. Approximately one-fortieth of the static weight on a wheel will therefore give the weight of the reciprocating parts which could be balanced without causing the wheel to rise from the track at diameter speed. This amount of balance would also double the load on the rail when the balance is down.

Common Methods.—Considering only the case of two-cylinder locomotives, each driving wheel should always have sufficient weight added to exactly counterbalance the crank pin, crank-pin hub and the part of the total weight of the side rod supported by the pin. In addition to the above, the main crank pin should have approximately one-half the total weight of the main rod plus (for outside valve gears) two-thirds the weight of the eccentric arm, considered as acting at crank-pin distance. This gives as nearly a perfect balance for the revolving weight as can be calculated by any fixed rule.

In addition to the weight calculated as above, the effect of the forces from the following reciprocating parts must be considered: the piston head, rod and nut; cross-head key, pin and nuts; approximately one-half the total weight of the main rod, and arm and link fastened to cross-head for outside valve gear. One method of taking care of these weights is to balance about two-thirds of the total weight of the reciprocating parts. The M. M. Asso. method has been to leave unbalanced on each side a portion of the reciprocating parts equal to $\frac{1}{400}$ of the weight of the locomotive but some companies have used the fraction $\frac{1}{360}$ instead of the above, the latter value being the value first proposed by G. R. Henderson.

The M. M. Committee, of 1915, questions both of the above methods and concludes from tests that the lighter the reciprocating parts can be made, the better the results obtained; and further, that when counterbalancing for very high speed, a larger per cent. of the reciprocating weights can be left unbalanced than has been the practice. In Table XXIV is shown some interesting ratios for certain locomotives built in 1914. The high figure in the last column in the first locomotive referred to in this table was obtained by the use of heat-treated steel in

the reciprocating parts, electric cast steel for the cross-head and by using hollow piston rods and crank pins. With a refinement of design along these lines, it is possible to construct reciprocating parts on each side $\frac{1}{240}$ part of the total weight of the locomotive in working order, instead of $\frac{1}{160}$ part, the latter value being considered a fair average.

TABLE XXIV.—WEIGHTS RECIPROCATING PARTS—LOCOMOTIVES BUILT DURING 1914

Road	Type	Cylinders	Drivers	Pressure	Piston load area × boiler pressure	Weight		
						Cross-head	Piston and rod	Main rod
1. Penna.....	4–4–2	23½ × 26	80	205	89,000	313	408	720
2. Penna.....	2–8–2	27 × 30	62	205	117,400	505	532	939
3. Penna.....	4–6–2	27 × 28	80	205	117,400	480	520	930
4. C. B. & Q....	2–10–2	30 × 32	60	175	123,700	526	945	1,035
5. P. & R.....	4–4–0	21 × 24	68½	220	76,200	302	464	590

Road	Type	Total weight reciprocating parts	Piston load carried				Weight in running order	Weight locomotive ÷ weight reciprocating parts
			Per pound of reciprocating parts	Per pound weight of cross-head	Per pound weight of piston	Per pound weight of main rod		
1. Penna.....	4–4–2	1,045	85	284	218	124	240,000	230
2. Penna.....	2–8–2	1,460	80	232	221	125	319,000	218
3. Penna.....	4–6–2	1,419	83	245	226	126	309,000	210
4. C. B. & Q...	2–10–2	1,936	64	235	131	120	370,000	191
5. P. & R.....	4–4–0	1,031	74	252	164	129	173,490	168

Counterbalance Rules.—A simple counterbalancing rule expressed in general terms, which should give good average results when applied to any class of locomotives in any service, has been stated as follows:

Rule 1.—*Keep the total weight of the reciprocating parts on each side of the locomotive below* $\frac{1}{160}$ *part of the total weight of the locomotive in working order, and then balance one-half the weight of the reciprocating parts.*

The above is based upon diameter-speed, and should keep the dynamic augment well within the limits of good practice. Where the normal speed is regularly considerably below the diameter-speed, it may be desirable to increase the proportion of the reciprocating weights to be balanced, to as much as 60 per cent. or 65 per cent.

Another counterbalancing rule is:

Rule 2.—Set an arbitrary percentage which the dynamic force of the overbalance will be allowed to increase the static weight.

For example: If it is desired that the dynamic force of the overbalance at a speed in miles per hour equal to the diameter of the driving wheel in inches, should not increase the static weight on a wheel more than 50 per cent., calculation could be made as follows:

4–4–2 type locomotive with 26-in. stroke.

Given: Static weight on one wheel = 30,000 lb.

To find: Maximum permissible weight of reciprocating parts to be balanced in one wheel = W.

$$W = \frac{50 \text{ per cent. static weight on one wheel} \times 0.312}{\text{crank radius in inches}} \qquad (99)$$

$$W = \frac{15{,}000 \times 0.312}{13} = 360 \text{ lb.}$$

Therefore, the total reciprocating weight to be balanced on one side of this locomotive would be 720 lb. And with 50 per cent. of the total reciprocating parts balanced on one side, the total weight of these parts must be designed to weigh 1440 lb.

Problem 31.—What should be the allowable static load on each wheel for the 2-8-2 locomotive in Table XXIV? Calculate the dynamic augment of the same and give the percentage of the value found to the static load on the wheel.

Problem 32.—Check each of the locomotives noted in Table XXIV by Rule 1 and Rule 2 above.

CHAPTER XV

ELECTRIFICATION OF STEAM RAILWAYS

Introductory.—The subjects, briefly discussed in this and the following chapter, are selected from many which might consistently be included in a study of locomotive operation and design. Among the other subjects referred to, the following may be listed: proper design of front ends, lubrication, roller and ball bearings, springs and equalizers, compound compressors, stresses on axles, tender construction, forced fits, tolerances for working parts, insulating against heat losses, the latest valve gears, safety first in design and operation, gasoline cars for light railroad service, draft rigging, economy in heating cars. The student should find these and like subjects worthy of careful study if he expects to pursue railroad work.

ELECTRIC LOCOMOTIVES

The electrification of steam railroads has progressed conservatively the past few years. It is one of the most important problems before the railroad engineer. The different systems adopted by leading railroads has supplied data from which a thorough study is being made of the application of steam railroad principles to electric locomotives. Future development must be determined by the co-operation of electrical engineers with men familiar with steam-locomotive design.

Like the throttle lever of the steam locomotive, the control lever of the electric locomotive may be placed in any one of its numerous notches to maintain the required speed. This difference, however, may be noted: the ability of the steam locomotive to maintain its speed continuously with heavy loads depends upon the capacity of the boiler; on the other hand, the electric locomotive has an ample supply of energy available, drawn from a large power house, and the limit of its endurance is determined by the safe temperature of the motor.

The best form of electric locomotive so far developed is one in which the running gear, frame and side rods are approximately the

same as on the steam locomotive. There is now a trend to develop electric locomotives which are free from reciprocating parts.

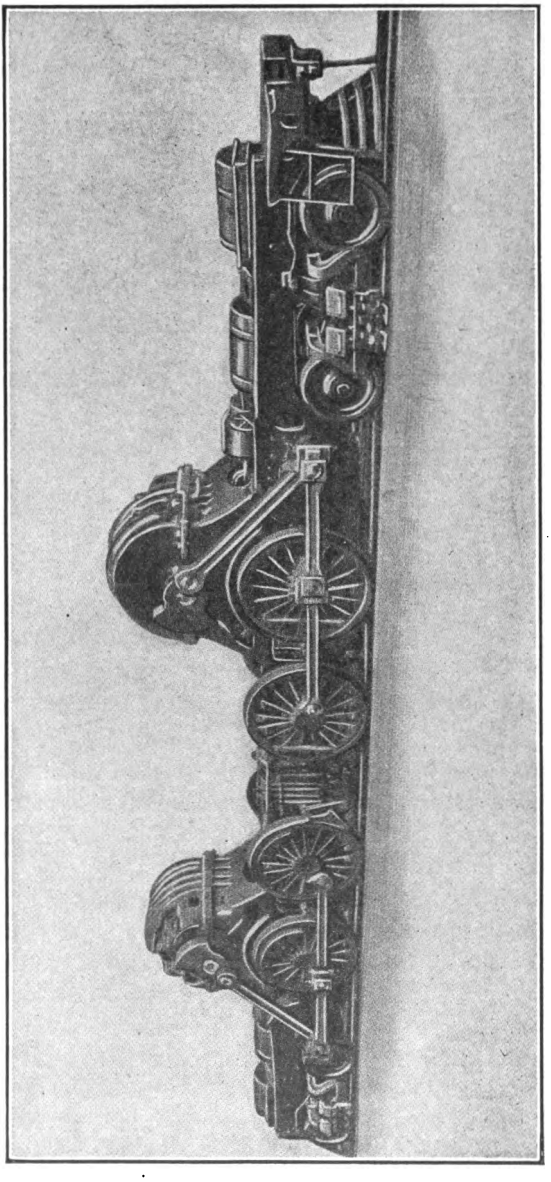


FIG. 112.—Under frames, motors and driving mechanism of Pennsylvania Railroad double articulated locomotive, used in New York tunnel service.

The earlier types of electric locomotives had their armatures concentric with the driving wheel axle and attached to driving

wheels by means of quills, which formed part of the armature. This construction of necessity causes a low center of gravity, and opinions differ among some of the leading locomotive designers on this point. Practically all of the N. Y. C. and N. Y., N. H. & H. locomotives are of this type and have met exacting require-

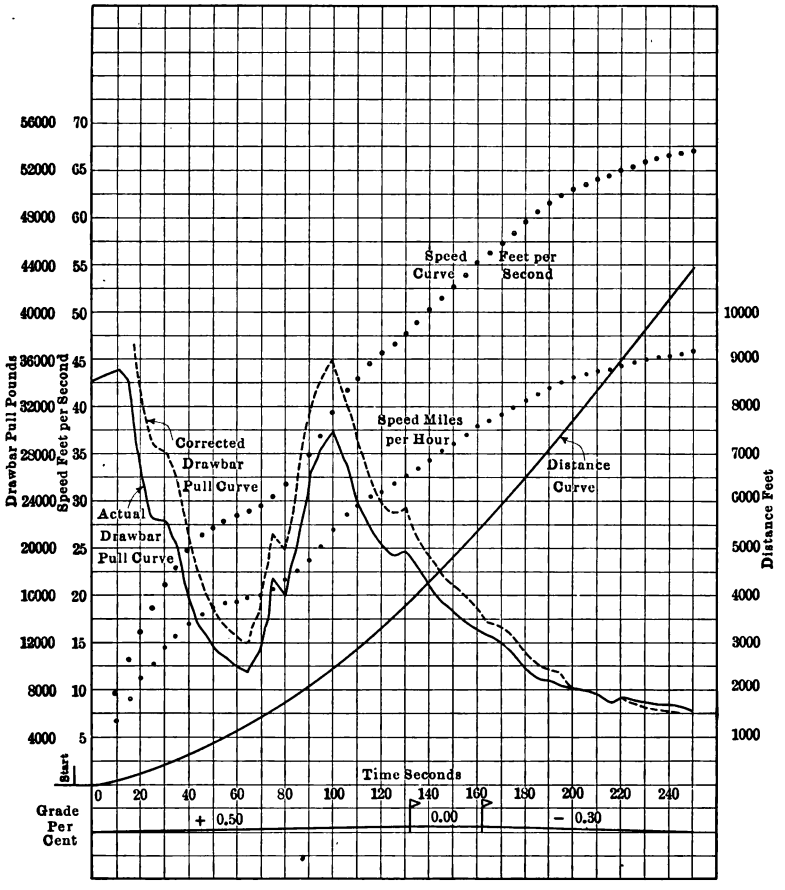


FIG. 113.—Starting curves for electric locomotive hauling 10 steel cars weighing 671 tons.

ments, whereas the ones with higher center of gravity, as operated by the P. R. R. in New Jersey and doing similar service, are entirely satisfactory.

Fig. 112, showing the driving mechanism of the Pennsylvania Railroad locomotive used in the New York Tunnel service, il-

lustrates the steam railroad principles applied to electric locomotives. The arrangement of weight, frames and general make-up of running gear is practically equivalent to two American type locomotives coupled back to back. The center of gravity of these locomotives is high, as the motor is mounted well above the driving axles. The transmission from motor to wheels is by crank and connecting-rod and these parts are protected from possible damage due to short circuit by interposing between the armature and its shaft a friction clutch which will slip before damaging stresses are imposed on the transmission.

Fig. 113 shows characteristic curves of one of the direct-current locomotives shown in Fig. 112, during a start with 10 cars weighing 671 tons, on the grade shown at the bottom of the illustration. It is of interest to note the character of the drawbar pull curve, comparing this with the results in Fig. 18, for steam conditions. These electric locomotives are designed for 550 tons trailing load and with this load the balancing speed on the level is 60 m.p.h. The maximum tractive power is 69,300 lb. Since an electric locomotive may safely develop for a short time an output which far exceeds its normal continuous capacity, the power and speed characteristics of electric locomotives differ from those of steam, as brought out by comparing Figs. 112 and 18.

COMPARISON OF SYSTEMS OF ELECTRIFICATION[1]

The salient features of the three systems of electrification for operating railroads are presented in diagrams Figs. 114 to 117, so arranged as to permit of a ready comparison between their essential characteristics. But one group of units are shown and auxiliaries such as switchboard apparatus are altogether omitted.

Fig. 114, showing the direct-current system, illustrates the alternating-current generator, the three raising transformers, the three-phase transmission circuit, the three sub-station lower transformers, and the rotary converter which supplies direct current to the third-rail contact system.

Fig. 115, illustrating the three-phase system, is similar to Fig. 114 up to the point where the power passes the sub-station transformers. Power is then delivered directly to the contact system, consisting of two overhead trolley wires, shown suspended from

[1] "Electrification of Railways," by George Westinghouse, Trans. A. S. M. E., 1910.

supporting cables in accordance with the commonly used catenary construction.

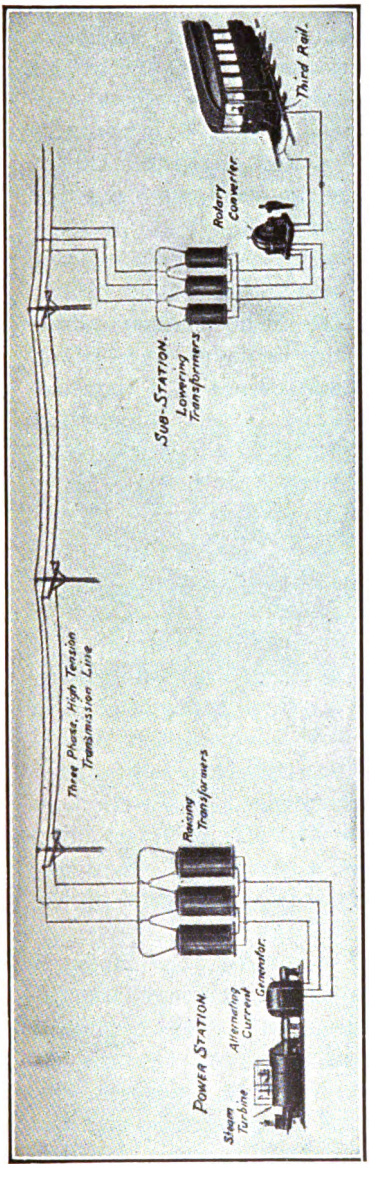


FIG. 114.—Direct-current railway system.

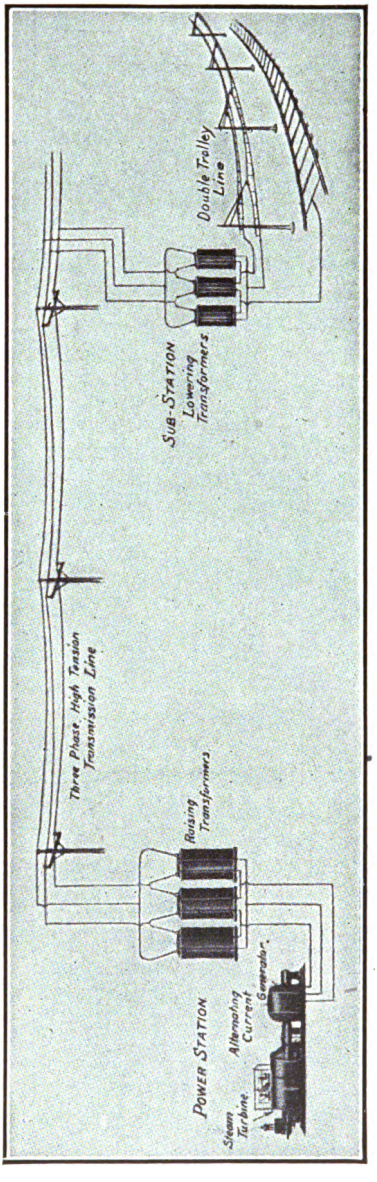


FIG. 115.—Three-phase railway system.

Fig. 116, presenting the single-phase system, has a similarity to the preceding sketch of the three-phase system, Fig. 115, and

may be derived from it by simplifying its several elements. Single transformers instead of groups of three are found in the power house and sub-station. The transmission has two wires instead of three and there is but one trolley wire instead of two.

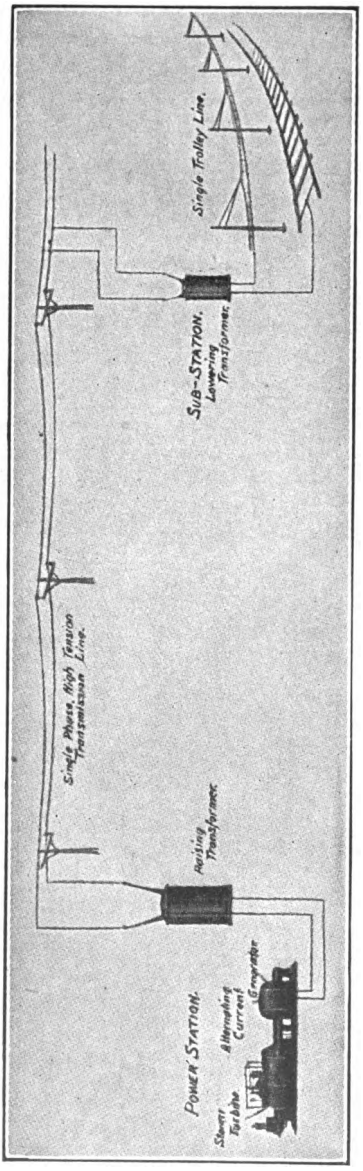


Fig. 116.—Single-phase railway system.

Fig. 117 shows the single-phase system where the distances are moderate and the generator can supply current directly to the trolley wire at 11,000 volts, thereby eliminating the high-tension transmission circuit and the sub-stations. This is the method employed in the single-phase installation on the New Haven system.

Table XXV gives a summing of important data regarding some locomotives built for heavy railroad service.

The Three Types of Motors.—The three types of electric motors have certain fundamental differences in speed performance which are important factors in determining the advantages, disadvantages and limitations of the several systems.

The Direct-current Motor.—The characteristics of the direct-current series railway motor are well known. It automatically adjusts its speed in accordance with the load, running more slowly if the weight of the train be greater, or the grade steeper. The speed with a given load, however, is definite; it is dependent upon

the voltage applied to the motor and cannot readily be varied. It is true that the speed can be decreased by inserting a resistance in the motor circuit, but this is wasteful and is inadmissible except as a temporary expedient. It is true also that the motors may be connected in series, thus dividing the pressure between two motors and thereby reducing the speed one-half; or if among four motors, to one-quarter speed. As the system of current supply involves a fixed voltage, it is obvious that for emergencies no speeds much above the maximum speed determined in the construction of the motor can be obtained. Furthermore, on account of the high cost involved in maintaining a practically constant voltage throughout the system, the voltage supplied to the motors often decreases considerably at the end of long lines, at the time of heavy load, thereby further reducing the speed attainable. It often happens in railway

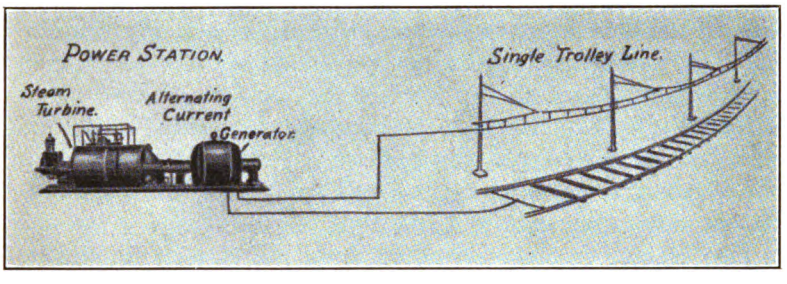


FIG. 117.—Single-phase railway without transmission system.

service that a locomotive should be operated somewhat above the normal speed, and sometimes a locomotive designed for freight service has to be pressed into passenger service. In such cases the speed with the direct-current locomotive would be considerably less than that necessary to maintain the schedule speed. However, for still further increasing the speed, a special form of field control can in certain cases be used.

The Three-phase Motor.—On the three-phase system, the motor is inherently a constant-speed motor; it runs at approximately the same speed at light load and at full load; it runs at nearly the same speed up a grade as on level track, although the horse power required on the grade may be several times that on the level. Conversely, it can run no faster on a level than it can climb a grade. In order to give a lower speed, however, the motors may be arranged upon the locomotive in pairs in a

manner equivalent to the arrangement of two continuous-current motors in series, just described. Motors may also be arranged for two or more speeds, but this involves some complication in windings and connections. In all cases lower speeds can be secured by the introduction of resistances which increase the losses and lower the efficiency. In no case can the speed in any of the arrangements of motors be appreciably higher at very light load than it is at full load.

The motors are of the induction type without commutators and their inherent limitations, and are of relative simplicity in construction. The current is usually supplied at 3000 volts from two overhead lines through two sets of current collectors.

With three-phase motors as now constructed and arranged upon locomotives, it is possible with no additional complication so to utilize the motors when locomotives are moving trains upon a descending grade that they become generators and return current to the line, a feature of value in certain mountainous districts but not of controlling importance in the selection of a universal system.

The Single-phase Motor.—The single-phase railway motor is a series motor with speed characteristics very similar to those of the direct-current motor, as the speed at a given voltage is greater or less, depending upon the load. The speed with a given load is also greater or less, depending upon the pressure applied to the motor; and this is not limited, as with direct-current motors, to that supplied by the circuit, and to one-half and one-fourth of that pressure, but is capable of adjustment to any desired degree of refinement by means of auxiliary connections from the secondary winding of the transformer on the locomotive, which is necessary for reducing the line voltage of 11,000 volts to the lower voltage required by the motors. Not only may numerous voltages less than the normal be arranged for lower speeds, but higher voltages can be provided to make possible speeds somewhat above the normal. In this simple manner a wide range of efficient speed adjustment is secured which is impossible with other systems. This facility of efficient operation over a wide range of speed and power requirements is one of the especially valuable features of the single-phase system.

The question of determination of the frequency for use on single-phase railways is one of very great importance. Twenty-five cycles is in general use for power transmission purposes and

has been adopted by nearly all the single-phase railroads now operating. A lower frequency permits of a marked reduction in the size of a motor for a given output, or conversely of a considerable increase in output from a motor of given dimensions and weight. Three-phase installations in nearly all cases employ approximately 15 cycles. The choice of frequency is one of the most involved, difficult and important problems now presented for solution.

Summary.—Locomotives equipped with each of the three types of motors have been in successful operation and have demonstrated their usefulness, capacity and reliability in practical railway service. The three-phase motor, having a definite constant-speed characteristic, is particularly adapted to conditions already mentioned; but on the other hand, it has a less general adaptability to the ordinary varying conditions of railway operation. The single-phase motor has a facility of voltage control which gives an efficient means of speed adjustment, and is in this particular superior to other systems.

The limitation of the original 600-*volt D.C. system was in the carrying capacity of the current-collecting parts. This made heavy train operation difficult. The higher trolley voltage of the A.C. systems makes it possible to deliver large amounts of energy to the locomotive without correspondingly large currents, which would overtax the current-collecting devices or parts.*

However, higher D.C. voltages, upward to 3000 *volts at the trolley, are rapidly coming into use. These systems are proving very successful in heavy train and trunk-line service and are growing so rapidly in favor among electrical engineers that current practice is tending strongly in that direction.*

TABLE XXV.—DATA ON ELECTRIC LOCOMOTIVES OF AMERICAN DESIGN

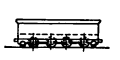
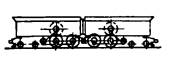
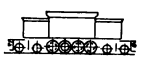
					
Built for	New Haven	Pennsylvania	New Haven	N. Y. C. & H. R. R.	Great Northern
Built by	Westinghouse	Westinghouse		General Electric	General Electric
Electric system	A.C., D.C.	D.C.	A.C., D.C.	D.C.	3-phase
Service	Passenger	Passenger	Frt. & pass.	Passenger	Frt. & pass.
No. motors per locomotive	4	2	2	4	4
Armature diameter, in	39½	56	76	29	35¾
Core length, including vent opening, in	18	23	13	19	16¼
Weight one motor, lb	16,420	45,000	41,600	18,150	15,000
Weight all motors on locomotive	65,680	90,000	83,200	72,600	60,000
Weight all electrical parts	110,400	127,200	135,000	91,200	109,000
Weight all mechanical parts	94,100	204,800	125,000	138,800	121,000
Weight complete locomotive	204,500	332,000	260,000	230,000	230,000
Weight on driving wheels	162,000	207,800	180,000	141,000	230,000
Weight complete locomotive for A.C. operation	196,000	D.C.	240,000	D.C.	230,000
Max. guar't'd speed, miles per hr.	about 86	about 80	45	75	30
Feature limiting speed	track	connecting rod	armatures	track	armature
Max. tractive effort	19,200	69,300	40,000	47,000	77,000
Loco. wt. in excess of 18% adhesion max. T.E., A.C. operation	88,700	none	17,500	none	none
Designed for trailing load, tons	250	550	1500 freight 800 pass.	.....	500 on 2.2% grade.
Balance speed on level with above load	about 75	60	35 freight 45 pass.	435 63	15

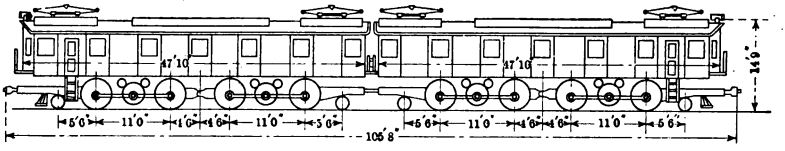
TABLE XXV*a*.—DATA ON LATEST ELECTRIC LOCOMOTIVES
NORFOLK & WESTERN RAILWAY

11,000 Volts A.C.

Weight total..... 270 tons
Weight on drivers..... 220 tons
Tractive effort.....110,000 lb. at 25 per cent. friction

Total horse power from eight motors, 6700
Tractive effort in per cent., 25

PRINCIPAL DIMENSIONS AND WEIGHTS



Length over all	105 ft. 8	in.
Driving wheel base	83 ft. 10	in.
Rigid wheel base	11 ft. 0	in.
Truck wheel base	16 ft. 6	in.
Height rail to pantograph—locked	16 ft. 0	in.
Height rail to top of cab—maximum	14 ft. 9	in.
Width over all—maximum	11 ft. 6¼	in.
Width over cab body	10 ft. 3	in.
Diameter of driving wheels	62	in.
Diameter of pony wheels	30	in.
Weight on drivers	220	tons
Total weight of locomotive	270	tons
Voltage of locomotive	11,000	volts a.c.
Number of motors	8	

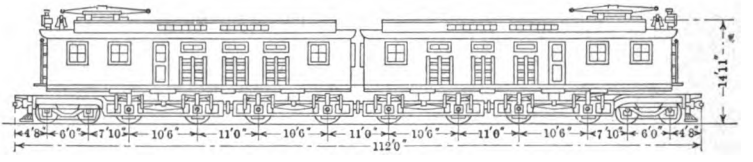
Locomotive performance under varying load conditions	Train on 2 per cent. grade	Train on 1 per cent. grade	Train on 0.4 per cent. grade
Weight of train-tons	3,250	3,250	3,250
Locomotives per train	2	1	1
Approximate speed, m.p.h.	14	14	28
Drawbar pull per locomotive, pounds			
Uniform acceleration	91,800	114,000	79,400
At speed on 2 per cent. grade	75,400	.....	.....
At speed on 1 per cent. grade	.....	85,000	.....
At speed on .4 per cent. grade	.....	.....	4,600
Maximum guaranteed accelerating tractive effort per locomotive	133,000	133,000	90,000
Approximate maximum guaranteed horse power developed by motors	5,000	5,000	6,700

TABLE XXV*a*.—DATA ON LATEST ELECTRIC LOCOMOTIVE (*Cont'd.*)
CHICAGO, MILWAUKEE & ST. PAUL RAILWAY

Weight total..... 260 tons
Weight on drivers..... 200 tons
Tractive effort.....85,000 lbs.

Total horse power from eight motors, 3440
Tractive effort in per cent. = 21.20
Each motor is twin geared to its driving axle

PRINCIPAL DIMENSIONS AND WEIGHTS



Total length of locomotive	112 ft.
Total weight	260 tons
Weight on drivers	200 tons
Weight on each guiding truck (2 four-wheel trucks)	30 tons
Number of driving axles	8
Total wheel base———103 ft.....Rigid wheel base	10 ft. 6 in.
Diameter of driving wheels	52 in.
Diameter of guiding wheels	36 in.
Number of motors	8
Horse power continuous rating	3,000
Voltage of locomotive	3,000 d.c.
Voltage per motor	1,500 d.c.
Horse power rating, 1 hour—each motor	430
Horse power rating, continuous—each motor	375
Horse power rating, 1 hour—complete locomotive	3,430
Trailing load capacity, 2 per cent. grade	1,250 tons
Trailing load capacity, 1 per cent. grade	2,500 tons
Approximate speed at these loads and grades	16 m.p.h.
Tractive effort—continuous rating	71,000 lb.
Per cent. of this tractive effort to weight on drivers	17.75
Speed at this tractive effort at 3000 volts	15.75
Tractive effort—1 hour rating	85,000 lb.
Per cent. of this tractive effort to weight on drivers	21.20
Speed at this tractive effort at 3000 volts	15.25 m.p.h.

CHAPTER XVI

MATERIALS OF CONSTRUCTION

The selection of suitable materials, with their accompanying specifications, is a subject that needs careful consideration, combined with sound judgment obtained from years of experience. The peculiarities incident to locomotive design, require that the materials selected for the various details fulfill the *analytical analysis* and that the *physical properties* in question agree with the *specifications*.

The character of a specification depends largely upon whether the material to be used is subjected to tension, compression, bending, torsion, shear (or a combination of these), wear, shock or expansion.

Ordinarily, the following materials are used to satisfy the above conditions:

TABLE XXVI

Tension Expansion	Steel { Carbon / Alloy Wrought iron	Compression	Steel { Carbon / Alloy Cast steel Cast iron Brass Bronze	Wear	Steel { Carbon / Alloy Cast iron Bronze Brass Babbitt and similar brands.
Bending Torsion Shear Shock	Steel { Carbon / Alloy Wrought iron				

Owing to the endless varieties of steels and other materials used in the construction of locomotives and the varying practices in vogue by the manufacturers in their selection, it is not deemed advisable to elaborate on this subject other than calling attention to the method of selecting the material, after which, a suitable specification can be obtained to cover the requirements. Practically every manufacturing plant have specifications covering materials best suited for their purpose.

As a means of establishing a basis in selecting suitable

material, the locomotive may be divided into the following divisions: 1, boiler; 2, frame and attachments; 3, running gear; 4, cab and fixtures.

From the divisions noted, the student should analyze the parts of each division for conditions, arranging them in accordance with Table XXVI. Example: A driving wheel axle, in addition to supporting a certain known weight upon its journals, is also subjected to shock and to torsional strains imposed upon it by sanding. This means that the axle must be designed for bending, shock and torsion. Referring to Table XXVI, we find that we can use either a plain Carbon Steel or Alloy Steel; Wrought Iron is no longer used for axles.

Railroads are giving the closest attention to the use of special steels. The following structural steels are now available:

.10 to .20 Carbon steel.
.20 to .30 Carbon steel.
.30 to .40 Carbon steel.
.40 to .50 Carbon steel.
.90 to .10 Carbon steel, etc.
.20 to .30 Carbon, with 3.50 per cent. Nickel.
.30 to .40 Carbon, with 3.50 per cent. Nickel.
.20 to .30 Carbon, with 1.00 per cent. Nickel, .50 per cent. Chromium.
.30 to .40 Carbon, with 1.00 per cent. Nickel, .50 per cent. Chromium.
.40 to .50 Carbon, with 1.00 per cent. Nickel, .50 per cent. Chromium.
.15 to .25 Carbon, with 3.00 per cent. Nickel, .50 per cent. Chromium.
.25 to .35 Carbon, with 3.00 per cent. Nickel, 1.50 per cent. Chromium.
.45 to .55 Carbon, with 3.00 per cent. Nickel, 1.50 per cent. Chromium.
.15 to .25 Carbon, with 1.00 per cent. Chromium, .15 per cent. Vanadium.
.25 to .35 Carbon, with 1.00 per cent. Chromium, .15 per cent. Vanadium.
35 to .45 Carbon, with 1.00 per cent. Chromium, .15 per cent. Vanadium.
.40 to .50 Carbon, with 2.00 per cent. Silicon, 1.00 per cent. Manganese.

With the introduction of new alloy steels has come the development in heat-treating steels. The term "heat treatment," as commonly used, indicates a sequence of heating, quenching and drawing operations, with many variations to meet existing differences in critical temperatures of the steels treated and of results desired. An alloy steel responds much better to heat treatment than does plain carbon. What alloy shall be selected depends largely on the cost as a function of the results obtained. Larger percentages of costly alloying elements cannot be considered; for example, considerable amounts of tungsten or vanadium would add too much to the cost. Tungsten has little

effect unless present to the extent of several per cent. In contrast, vanadium need not be present except in very small amounts —0.15 to 0.25 per cent. to produce results.

Chromium, manganese and silicon are cheap elements to add. Chromium need be present to the extent of a per cent. or two. Manganese exercises much influence even in fractions of a per cent. and entirely alters the character of an alloy in several ranges of possible use. The cost of added manganese may hardly be considered. It occurs often as a natural alloy of iron and is cheap. Silicon is low in cost and powerful in influence in amounts of a per cent. or two. It is a natural alloying element for iron and therefore cheap in the shape of pig iron and ferro-silicon.

Nickel is a rather expensive alloying element and gives best service in structural steels containing from 1 to 5 per cent. Chromium and nickel are found in ore and in such form as to pass on through all the metallurgical steps to the finished steels. Such alloy steel is within reach for nearly all purposes. The response to heat treatment is very good with the nickel content at 1 per cent. and the chromium at 0.50 per cent., it being understood that the carbon content be suitably chosen.[1]

Some of the parts which have been heat treated to advantage are: driving axles, tires, connecting rods, crank pins, piston rods, and springs.

High-priced steel should not be used where cheaper steel or other metal may be used. It is often wise and economical to depend on a steel of low original cost and then treat it, as in case-hardening steel. Again, it is wise to buy a steel that is used in the condition purchased, and more often it is best to combine a medium cost of material with heat treatment.

It is thought by some that if a high-priced alloy steel be purchased and used as purchased, that full value is thereby obtained. This is a mistake, as there is no alloy steel that will yield anywhere near the benefit that will compensate for the increased cost, without intelligent heat treatment at some stage of its adaptation for use. An alloy steel in a natural condition, or in an annealed condition, is but little better than a good plain carbon steel. In other words, 15c. steel is but little better than

[1] During the past few years, the Proceedings of the American Society for Testing Materials, contain the recommendations for the proper heat treatment of steel for different purposes. Refer also to recent Proceedings A. R. M. M. A.

3c. steel, unless it be developed physically by suitable treatment, as illustrated by the following tests made by Mr. Henry Souther:

	Elastic limit per sq. in.	Fiber stress applied	Endurance
0.40 Carbon steel, "as received."	59,800 lb.	54,800	19,300 revs.
0.22 Carbon chrome-vanadium "as received" (1.00 per cent. chromium)		54,800	46,900 revs.
0.25 Carbon, 3.50 per cent. nickel, "natural"	54,000 lb.	54,800	75,400 revs.
0.30 Carbon, 3.50 per cent. nickel "heat treated"	81,100 lb.	54,800	10,814,000 (did not fail)

Along the same line of reasoning, steels must be chosen in relation to the design, proportions and use of any given part. If a 3c. steel will do a piece of work, then there is no reason for the use of a more expensive steel, whether the expense be in the original cost of material or the cost of treatment. *If a factor of safety of ten exists and suffices, then a material that will give a factor of twenty is of no greater value.* All forms of degrees of excesses are committed in the selection of steels along these lines. The poorest steel is put to uses and the best of results expected from it, where only the best should be used. Steels are used in a natural and uncertain state when the greatest certainty should prevail, and at other times are treated when a natural condition would answer all demands. In between these extremes lies the reasonable and economical course; the hardest to follow. Theory and practice must unite; laboratory tests and empirical knowledge must be used in combination.

Selection of the Steel to Use.—Continuing directly the application of Table XXVI, the following from Mr. Souther brings out the methods of studying the conditions before selecting a method for a particular service.

In the driving axle of a locomotive is found a part that might well be made of a thoroughly heat treated steel. It is of large dimensions and has to be forged with several heatings, and if left in a natural condition, it is, necessarily, in a non-homogeneous condition. Homogeneity should exist in such an important part; in fact, it should exist in any axle.

It would seem that a driving axle properly designed does not need to possess a very high elastic limit, but it does need to be in a condition to resist vibration, fatigue and impact. This means that it must be fine grained. Stiffness must be secured by large dimensions, and such dimensions as will yield the necessary stiffness will usually yield more than enough strength; but if there be certain designs where space or weight is limited, then the alloy steel may be widely resorted to in order to obtain a high elastic limit with very greatly increased powers of resistance to fatigue.

The connecting rod of the locomotive must be forged, and after forging naturally cannot be in a homogeneous condition. It should therefore receive some treatment after forging.

In such small parts as the wheel spindles of an automobile, alloy steels have been substituted for wrought iron and for 0.40 to 0.50 carbon steel, with the result that breakage ceased entirely. This was an instance where design could not be changed and where alloy steel was a great boon to the manufacturer of the automobile. Change of design would have resulted in complete redesign of axle and wheel.

The locomotive crank pin, like the axle, may be of such generous design, in order to get sufficient bearing surface, as to possess surplus strength. This being the case, there is no need of resorting to an alloy steel. At the same time, it must be borne in mind that a crank pin forged from an alloy steel of the proper composition, and properly heat treated, is, without question, much more reliable than steel in any other condition.

If a connecting rod must be stiff, to withstand the "whip" of rapid reciprocating movement, possibly the design demands so much material that strength is not an important consideration. Under such condition, alloy steel is not necessary. On the other hand, if fatigue due to vibration be an important factor, then the fine grain of an alloy steel resulting from heat treatment becomes important.

The piston rod is a part to which the remarks on connecting rods also apply; possibly to a less extent as to severe punishment. There is no question but what the piston rods should be made of homogeneous material, and therefore bettered by heat treatment. The problem is to ascertain whether or not a material of very high elastic limit is necessary.

The frame of a locomotive presents very different problems. It seems as if an easily forged, easily welded steel would be a most important consideration. If an alloy steel is chosen, it must be with these qualities in mind.

The shape of the frame is irregular and will not permit easy handling for heat treatment. Annealing after forging is certainly desirable in order to bring the frame into a homogeneous condition with uniform grain throughout. That the sharp angles would withstand heat treatment is a question. If the treatment be a practical operation, the resulting benefit cannot be questioned. This problem is not unlike that encountered in the automobile frame, which is very thin, consequently, very sensitive to heat treatment. It is irregular in shape and difficult to handle. Nevertheless, heat treatment is carried on in a commercial way and with success.

The most reliable frames and the strongest frames to-day are those of heat treated alloy steel. As far as locomotive frames are concerned, it is a question whether or not the benefits received warrant the extra cost of the material and the cost of heat treatment.

APPENDIX

TABLE XXVII.—REVOLUTIONS PER MINUTE FOR VARIOUS DIAMETERS OF DRIVING WHEELS AND SPEEDS

Diameter of wheel	Miles per hour							
	10	20	30	40	50	60	70	80
50 in.	67	134	201	268	336	403	470	538
56 in.	60	120	180	240	300	360	420	480
60 in.	56	112	168	224	280	336	392	448
62 in.	54	108	162	217	271	325	379	433
66 in.	51	102	153	204	255	306	357	408
68 in.	49	99	148	198	247	296	346	395
72 in.	47	93	140	187	233	279	326	373
78 in.	43	86	129	172	215	258	301	344
80 in.	42	84	126	168	210	252	294	336
84 in.	40	80	120	160	200	240	280	320
90 in.	37	75	112	150	186	224	261	299

TABLE XXVIII.—PISTON SPEED IN FT. PER MIN. AT ENGINE SPEED OF 10 MILES PER HR.

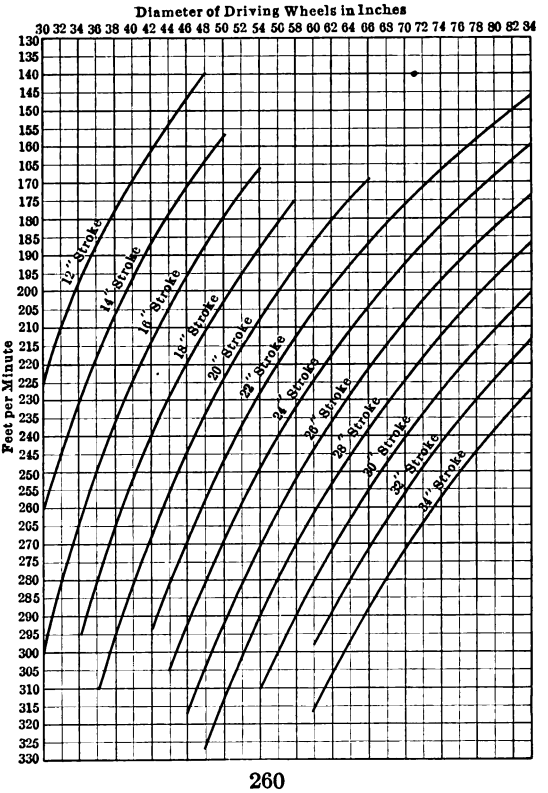


TABLE XXIX.—MILES PER HR. AND FEET PER SEC.

Miles per hr.	Ft. per sec.	Miles per hr.	Ft. per sec.	Miles per hr.	Ft. per sec.
10	= 14.65	43	= 63	72	= 105½
		44	= 64½	73	= 107
15	= 22			74	= 108½
16	= 23½	45	= 66		
17	= 25	46	= 67½	75	= 110
18	= 26½	47	= 69	76	= 111½
19	= 28	48	= 70½	77	= 112¾
		49	= 71¾	78	= 114¼
20	= 29¼			79	= 115½
21	= 30¾	50	= 73¼		
22	= 32¼	51	= 74¾	80	= 117
23	= 33¾	52	= 76¼	81	= 118¾
24	= 35¼	53	= 77¾	82	= 120¼
		54	= 79¼	83	= 121¾
25	= 36¾			84	= 123¼
26	= 38¼	55	= 80¾		
27	= 39½	56	= 82¼	85	= 124¾
28	= 41	57	= 83½	86	= 126¼
29	= 42½	58	= 85	87	= 127¾
		59	= 86½	88	= 129
30	= 44			89	= 130½
31	= 45½	60	= 88		
32	= 47	61	= 89½	90	= 132
33	= 48½	62	= 91	91	= 133½
34	= 50	63	= 92½	92	= 135
		64	= 94	93	= 136½
35	= 51¼			94	= 137¾
36	= 52¾	65	= 95¼		
37	= 54¼	66	= 96¾	95	= 139¼
38	= 55¾	67	= 98¼	96	= 140¾
39	= 57¼	68	= 99¾	97	= 142¼
		69	= 101¼	98	= 143¾
40	= 58¾			99	= 145¼
41	= 60¼	70	= 102¾		
42	= 61½	71	= 104¼	100	= 146½

TABLE XXX.—CIRCUMFERENCES AND AREAS OF CIRCLES

Diam.	Circum.	Area	Diam.	Circum.	Area
10	31.416	78.540	24	75.398	452.39
½	32.987	86.590	½	76.969	471.44
11	34.558	95.033	25	78.540	490.87
½	36.128	103.87	½	80.111	510.71
12	37.699	113.10	26	81.681	530.93
½	39.270	122.72	½	83.252	551.55
13	40.841	132.73	27	84.823	572.56
½	42.412	143.14	½	86.394	593.96
14	43.982	153.94	28	87.965	615.75
½	45.553	165.13	½	89.535	637.94
15	47.124	176.71	29	91.106	660.52
½	48.695	188.69	½	92.677	683.49
16	50.265	201.06	30	94.248	706.86
½	51.836	213.82	½	95.819	730.62
17	53.407	226.98	31	97.389	754.77
½	54.978	240.53	½	98.960	779.31
18	56.549	254.47	32	100.531	804.25
½	58.119	26 .80	½	102.102	829.58
19	59.690	283.53	33	103.673	855.30
½	61.261	298.65	½	105.243	881.41
20	62.832	314.16	34	106.814	907.92
½	64.403	330.06	½	108.385	934.82
21	65.973	346.36	35	109.956	962.11
½	67.544	363.05	½	111.527	989.80
22	69.115	380.13	36	113.097	1017.9
½	70.686	397.61			
23	72.257	415.48			
½	73.827	433.74			

TABLE XXXI.—CYLINDER VOLUMES IN CUBIC FEET FOR ONE CYLINDER

Diameter, in.	Stroke in inches							
	22	24	26	28	30	32	34	36
16	2.55	2.78	3.02	.....	.....	.....	.....	.....
16½	2.72	2.97	3.23	.....	.....	.....	.....	.....
17	2.90	3.17	3.43	3.69	.....	.....	.....	.....
17½	3.06	3.34	3.72	3.98	.....	.....	.....	.....
18	3.23	3.53	3.82	4.12	.....	.....	.....	.....
18½	3.42	3.73	4.04	4.35	.....	.....	.....	.....
19	3.61	3.94	4.26	4.59	4.92	.....	.....	.....
19½	3.80	4.15	4.50	4.84	5.19	.....	.....	.....
20	4.00	4.37	4.73	5.10	5.46	.....	.....	.....
20½	4.20	4.58	4.96	5.34	5.73	.....	.....	.....
21	4.40	4.80	5.20	5.60	6.00	6.42	.....	.....
21½	4.62	5.04	5.46	5.88	6.30	6.72	.....	.....
22	4.84	5.28	5.72	6.16	6.60	7.05	.....	.....
22½	.....	5.52	5.98	6.44	6.90	7.36	.....	.....
23	.....	5.76	6.24	6.72	7.20	7.68	8.18	.....
23½	.....	6.02	6.52	7.02	7.52	8.02	8.52	.....
24	.....	6.27	6.79	7.31	7.83	8.38	8.90	.....
24½	.....	.....	7.10	7.65	8.20	8.75	9.30	.....
25	.....	.....	7.38	7.95	8.52	9.08	9.65	10.21
25½	.....	.....	7.68	8.27	8.86	9.45	10.04	10.63
26	.....	.....	7.98	8.60	9.21	9.83	10.44	11.08
27	.....	.....	8.61	9.27	9.93	10.59	11.26	11.92
28	.....	.....	9.26	9.97	10.68	11.39	12.10	12.82
29	.....	.....	9.93	10.70	11.46	12.22	13.00	13.76
30	.....	.....	10.63	11.45	12.27	13.09	13.90	14.72
31	.....	.....	.....	12.23	13.10	13.97	14.84	15.71
32	.....	.....	.....	13.03	13.96	14.90	15.84	16.76
33	.....	.....	.....	13.85	14.84	15.82	16.82	17.81
34	.....	.....	.....	14.71	15.76	16.82	17.87	18.92
35	.....	.....	.....	15.59	16.70	17.81	18.92	20.10
36	.....	.....	.....	16.49	17.67	18.85	20.03	21.21

TABLE XXXII.—HEATING SURFACE OF BOILER TUBES AND SUPERHEATER FLUES

Length in ft.	Heating surface in sq. ft. — Outside diameter						
	1¾″	2″	2¼″	2½″	5″	5¼″	5½″
10	4.582	5.236	5.891	6.545	13.090	13.744	14.399
11	5.040	5.760	6.480	7.199	14.399	15.118	15.839
12	5.498	6.283	7.069	7.854	15.708	16.493	17.279
13	5.956	6.807	7.658	8.508	17.017	17.867	18.719
14	6.414	7.330	8.247	9.163	18.326	19.242	20.159
15	6.872	7.854	8.836	9.817	19.635	20.616	21.599
16	7.320	8.378	9.425	10.472	20.944	21.990	23.038
17	7.788	8.901	10.014	11.127	22.253	23.365	24.478
18	8.246	9.425	10.603	11.781	23.562	24.739	25.918
19	8.705	9.948	11.192	12.435	24.871	26.114	27.358
20	9.164	10.472	11.781	13.090	26.180	27.488	28.798
21	.....	10.995	12.360	13.744	27.489	28.862	30.238
22	.....	11.519	12.959	14.398	28.798	30.236	31.678
23	.....	12.043	13.549	15.053	30.107	31.610	33.118
24	.....	12.566	14.138	15.708	31.416	32.984	34.558

TABLE XXXIII.—EFFECT OF IMPURE WATER IN BOILERS

	1	2	3	4
Trouble.....	Soft scale	Hard scale	Corrosion	Foaming
Cause.....	Lime carbonate, magnesia carb'nt	Lime sulphate, magnesia sulphate	Acids, chlorides	Alkali, mud
Care of boilers	Through wash-outs frequently	Through wash-outs frequently	Close inspection frequently	Blow out and change water frequently
Remedy	Slaked lime	Soda ash with or without slaked lime	Slaked lime or soda ash	Distillation, alum
Possible after trouble	Should be none	Foaming	Foaming	Some corrosion if all distilled water be used
Boiler treatment	Ordinary	Blow out and change water	Blow out and change water	Mixture with other waters

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